P

INJECTORS:

THEIR

THEORY, CONSTRUCTION, AND WORKING.

BY

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THIRD EDITION.

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PREFACE TO FIRST EDITION.

In this work it has been thought advisable to stray somewhat from the well-trodden path so generally taken under similar circumstances, and to reserve the historical portion until the theory has been discussed at some length. With this arrangement, it is the Author's opinion, the general reader is placed in a more fitting position to follow and appreciate the different stages of development; and carrying out this idea, this book *concludes* with an historical summary of the Injector and cognate apparatus.

The first chapter opens with a popular explanation of the action of the steam injector. Immediately following will be found the mathematical investigation of the velocity of efflux of the steam jet; the different steps throughout this and the remainder of the work being given with much consecutiveness, more especially for the purpose of assisting those who may not be quite familiar with the mathematics there used in following the reasoning.

To permit those who are not able to follow the latter part of Chapter I., a table has been constructed at the end of Chapter II., giving the principal data and results which may be needed in any of the calculations that come after.

The method of treatment given in Chapters VI. and VII. is due to Professor Elliott, D.Sc., much of it being taken from his paper there mentioned, which he kindly placed in the Author's hands.

In the rather large number of illustrations which follow, the Author has endeavoured to give examples of the different forms rather than examples of each manufacturer's apparatus. The generosity of makers and patentees in supplying blocks for the illustrations, where possible, has rendered the preparation of this work much less arduous than it otherwise would have been, and the Author desires to express his thanks to them.

It will, no doubt, be noticed, that of the examples given very few are of American origin, which fact is due to the difficulty of obtaining them, together with the impossibility of consulting any American patent records.

Although the exhaust injector is identically similar to the ordinary live-steam apparatus, except in dimensions, it has been thought desirable to treat it separately, together with compound injectors. The same applies to the ejector condenser.

In the historical portion, the dates given have been almost exclusively taken from the patent specifications, as have also many of the illustrations ; and it may here be remarked that much useful information may be obtained from the volume of abridgments of patent specifications (Injectors) recently published by the Patent Office.

The only treatise published in this country in which the Author has been able to find more than a passing reference to the injector is Professor Peabody's admirable work on Thermodynamics, which the Author read with much interest soon after its publication.

To endeavour to render any portion of this book readable at any time, the different equations have all been numbered consecutively, and copious references have been made to them throughout. Also an index of symbols has been made, so that the reader may, by referring to it, interpret any equation or formula, without having to first wade through the previous pages to find out what each symbol represents.

University College, W. W. F. P.
Cardiff, 1893.

PREFACE TO SECOND EDITION.

ADVANTAGE has been taken in the preparation of this edition to describe the latest patterns of injectors, and the editor desires to thank Mr. William Dean, Messrs. Gresham and Craven, Messrs. Holden and Brooke, Mr. James Holden, Messrs. Schäffer and Budenberg, The Exhaust Injector Company, and Mr. F. W. Webb for their kind assistance.

Manchester,
September, 1900.

PREFACE TO THIRD EDITION.

SINCE the first edition of this work appeared much experimental work has been done on the steam jet (principally on account of its importance in connection with steam turbines), and the Author has, in consequence, felt it necessary to re-write a good part of the first half of the book, putting in some new matter and simplifying some of the old.

Many new illustrations have been added, for which the Author desires to thank the different makers.

London, W. W. F. P.
October, 1905.

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THE INJECTOR.

CHAPTER

INTRODUCTION.

...N injector is one of the simplest ... ngineer has invented, but its ... ess than a paradox to many of ... s to use it.

Before going into details it may ... ery simply how it works and ... roducing the *apparently* paradoxical ...

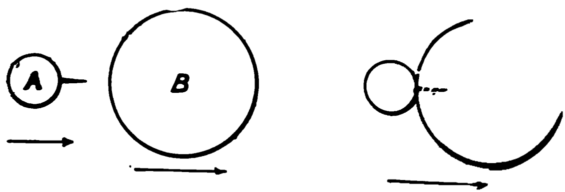


FIG. 1.

...team from a boiler and forcing many times ... f feed water back into the same boiler, ... oiler at a higher pressure.

Assume for simplicity that we have two bodies ... 3, fig. 1, whose motions are not resisted in ... hat is, they are perfectly free to move.

Now, let A be endowed with motion in the ... owards B. Further, let B be composed of ... naterial, such as wood, so that a spike in A ... 3 and hold the two bodies together after A ...

B. Whether B has any initial motion or not, the principle of the "Conservation of Momentum" (derived from the second and third laws of motion) states that the total momentum of A and B is unchanged by the collision, or, in other words, "*The momentum of A before impact,*

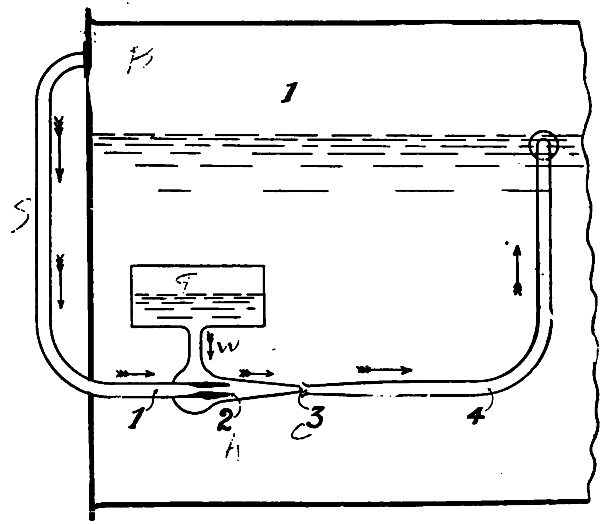


FIG. 2.—Diagram of Injector Arrangement.

plus the momentum of B before impact, must equal the momentum of A and B moving together as one body after impact."

Had the masses of A and B been 1 and 9 lb. respectively, and their original velocities in the direction of the arrows been 2,000 and 25 ft. per second respectively, the sum of their momenta before impact would have been

$$1 \times 2{,}000 + 9 \times 25, \text{ or } 2{,}225 \text{ units.}$$

After impact the two bodies moved together with some velocity V ft. per second, and their momentum was

$$(9 + 1)\ V, \text{ or } 10\ V \text{ units.}$$

As these two results must be the same, we have

$$2{,}225 = 10\,V,$$
$$\text{or} \qquad V = 222{\cdot}5 \text{ ft. per second.}$$

A general idea of the injector can be obtained from the diagrammatic sketch fig. 2.

Steam passes from the boiler B by the pipe S, and leaves the latter near A at a velocity of, say, 2,000 ft. per second. This velocity corresponds to a boiler pressure of about 40 lb. per square inch by the gauge. At A it would overtake water flowing from the tank T by the pipe W at a velocity of, say, 25 ft. per second, corresponding to a head of about 10 ft. The cold water and the steam mix and move together after impact much in the same manner as the hypothetical bodies A and B in fig. 1. This being the case, the final velocity of the mixture of steam and water leaving C must be 222·5 ft. per second if there be 9 lb. of water to 1 lb. of steam—a very common proportion.

We now desire to see what will happen to the stream after it leaves the orifice C. The height of a column of water which would produce the same pressure as that in the boiler is 2·3 times the pressure there is in pounds per square inch, or 92 ft., assuming the pressure to be 40 lb. per square inch, as before suggested. The velocity V with which water would flow out of the boiler is given by

$$\frac{v^2}{2g} = 92 \text{ or } v = 77 \text{ ft. per second.}$$

Conversely, if a stream of water were directed against the orifice near C in the opposite direction, with a velocity of 77 ft. per second, that stream would just enter the boiler at a pressure of 40 lb. per square inch, and hence a stream leaving C at 222·5 ft. per second in the same manner would not only enter the boiler very easily, but, in addition, would overcome considerable frictional or other resistances on its way to the boiler, if such resistances existed.

This stream leaving C at 222·5 ft. per second and entering the adjoining orifice is the combined stream of

water and steam after they have collided and mixed and the steam condensed to water.

FIG. 3.—Section of Simple Non-lifting Injector.

It is evident from the above that not only will an injector easily feed its own boiler, but it will force water into a boiler at a higher pressure than that from which i

derives its steam, and, further, an injector using the steam from the exhaust pipe of an engine may force water into a boiler at a pressure of 40 or 50 lb. per square inch.

CHAPTER II.

Construction and Arrangement of Simple High-pressure Injectors.

An injector has three principal parts, namely, the steam cone; the combining cone where the steam collides with and is condensed by the water; and the delivery cone by which the kinetic energy of the combined stream is converted into pressure energy before entering the boiler or feed pipe.

These cones are seen in fig. 3, which is a not very recent design of a non-lifting type. The overflow gap between the combining and delivery cones is for the purpose of providing an outlet for steam and water at starting, until the combined jet is sufficiently established to enter the boiler. It also permits of an excess of water escaping to the atmosphere should such excess exist.

Types of Simple High-pressure Injectors.

Lifting injectors are designed to suck up their feed water from a lower level; while *non-lifting* injectors must be supplied with feed from a tank above the level of the injector.

There is no reason why the latter form should be made at all now, as the extra expense of making the lifting type is comparatively small, and there is the further advantage that the injector is then capable of re-starting itself should its action be momentarily stopped.

For an injector to be able to suck up its feed from a lower level, the steam jet must be arranged to have a pressure in the neighbourhood of the entrance to the combining cone, less than that of the atmosphere. This is produced by giving a suitable shape* to the steam cone,

* See chapter on the Steam Jet.

and at the same time providing a sufficiently large outlet for the steam through the overflow chamber into the atmosphere.

Referring for a moment to fig. 3, it will be noticed that the end of the combining cone next to the overflow is smaller than the end of the steam cone, and consequently the steam which leaves the steam cone will not be able to pass through the combining cone orifice without contracting in volume. The volume expands rather than contracts as the pressure falls, and therefore there is likely to be a choking of the combining cone by the steam

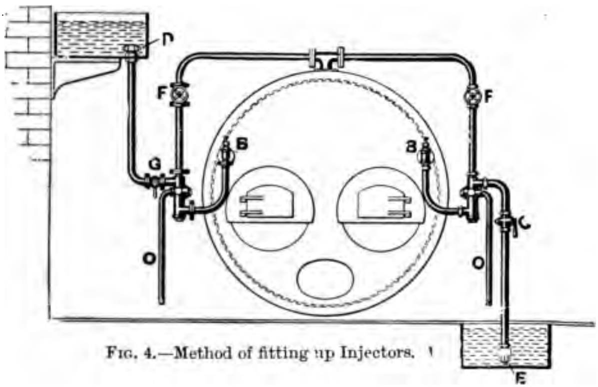


FIG. 4.—Method of fitting up Injectors.

unless condensed by the incoming feed, which choking will cause the steam to pass into the feed pipe and drive the feed away from the injector. Hence, with a non-lifting injector, the feed must be turned on first to condense the steam directly it reaches the combining cone, so as to prevent choking, and permit the combined jet to become established.

NON-LIFTING INJECTORS.

All non-lifting injectors have practically the same arrangement of cones, though the methods of construction may vary to some extent. In fig. 3 the whole of the injector is constructed of gun metal, and the different

parts are screwed together. Ribs strengthen the cones longitudinally.

It is generally more economical to use an outer cast-iron casing, as in fig. 5, where the gun-metal cones are held in position by the coupling flanges. The contours of the cones in fig. 5 are not so carefully designed as in fig. 3.

The general arrangement of pipes and fittings for a non-lifting injector is shown on the left of fig. 4. A cock in the water-supply pipe is necessary for the purpose of regulating the amount of water, so that there shall be no overflow.

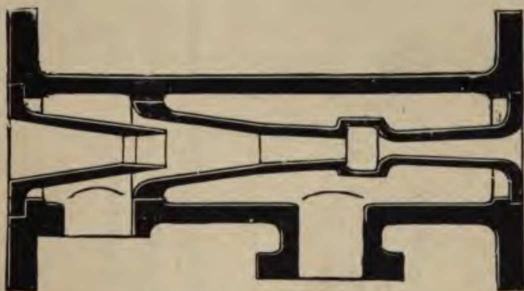


FIG. 5.—Section of Non-lifting Injector.

The same result can be accomplished by adjusting the aperture between the steam and combining cones to suit the required amount of water. An arrangement due to Mr. F. W. Webb, and used on the London and North-Western Railway, is shown in fig. 6, in which two sections are given.

It is fixed under the footplate at the back of the boiler. The steam cone points upwards, while the combining and delivery cones are cast in one piece, and made so as to be capable of sliding in the outer casing, and thus permitting of the adjustment of the water inlet space between the steam and combining cones, the adjustment being carried out by means of a vertical rod screwed at its upper end into the corresponding hand wheel W.

The delivery pipe terminates in the clack-box casting fixed to the back plate in the usual manner. The

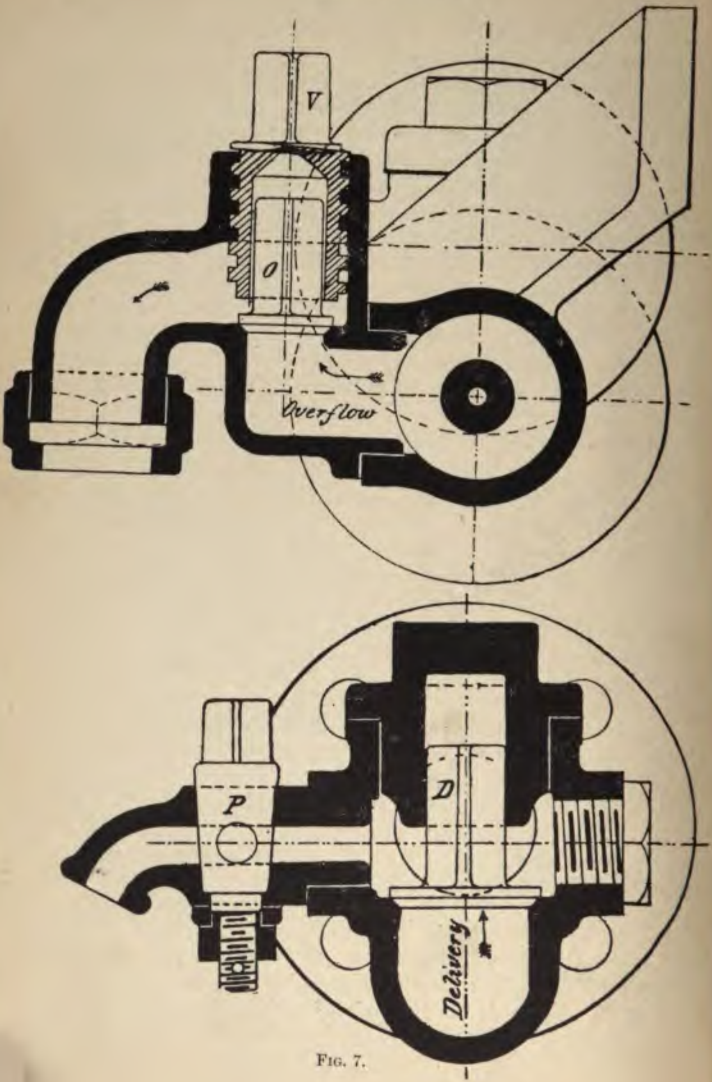


FIG. 7.

back-pressure valve has its seating on the end of the diverging cone, and is of the form generally used for that purpose, having a projection turned on the part facing the cone, to guide the stream lines without producing eddies, and thereby causing a resistance to the flow of water from the injector. The narrow portions of the cones are strengthened by ribs. The regulation of the water entering the combining cone is accomplished by moving the steam cone along its axis, and thus extending or contracting the annular space between it and the combining cone. Two lugs are cast, one upon the hollow rod near its upper end, and another on the outer casing adjoining the stuffing box. In the former of these a spindle turns freely, having on its lower half cut five or six separate threads of the same pitch; this screwed portion fitting into a similar screwed hole in the latter lug. By securing a washer on the screwed spindle just below the upper lug, the steam cone will receive the same axial motion as the screwed spindle. The object of the large number of threads on the water-regulating spindle is to complete the regulation instantly with only a partial turn of the handle.

We will assume it is required to start feeding a boiler by lifting the feed water from a well. In this form of apparatus there are no openings in the combining cone through which the steam can escape to the atmosphere, except through the usual overflow orifice, and therefore it will be absolutely necessary to use a steam orifice much smaller than the overflow during the time which elapses before the feed water reaches the combining cone. This is done by withdrawing the central spindle slightly by a partial turn of the hand wheel. The steam cone is then raised by turning the other handle. When the water appears at the overflow the hand wheel is turned back, so as to completely withdraw the plug spindle, thereby admitting a full supply of steam, or sufficient to drive the feed water into the boiler. Should either water or steam appear at the overflow after this, its particular regulator must be partly closed until it disappears. The illustration, fig. 8, represents an injector made by Messrs. Holden and Brooke. Most of the standard makers manufac

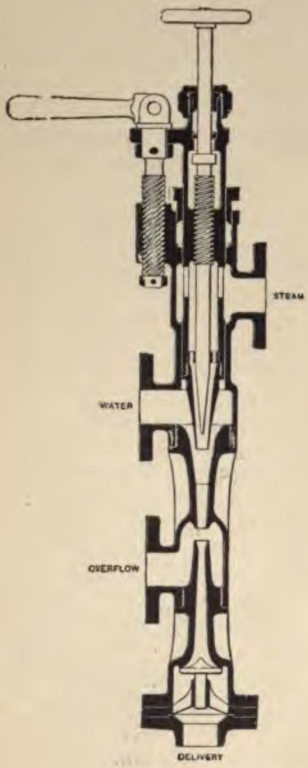


FIG. 8

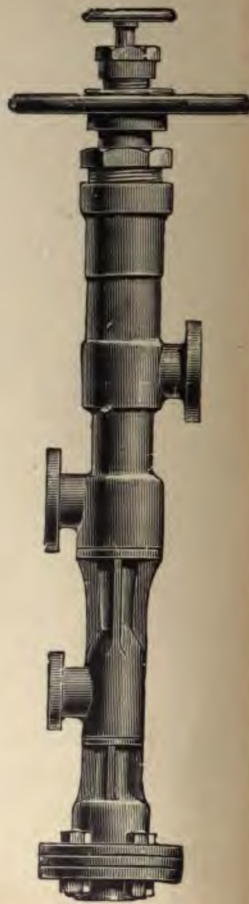


FIG. 9.

Giffard's Injector.

similar injector. This pattern is called the "eccentric," to distinguish it from a slightly different variety called the "concentric," fig. 9, now nearly obsolete, though at one time made by Messrs. Sharp, Stewart, and Company, who originally introduced the invention into England soon after it was brought out by M. Giffard. The above firm continued to carry on the manufacture of injectors without intermission for nearly a quarter of a century, until it was decided to remove their engineering works from Manchester to Glasgow, when they disposed of the whole of their injector business to the Patent Exhaust Steam Injector Co. Limited, of Manchester (now Messrs. Davies and Metcalf Limited).

The only difference between figs. 8 and 9 is the method of raising the steam cone. In the "concentric" pattern a larger hand wheel is keyed to the hollow spindle which carries the steam cone. A thread is cut upon this hollow spindle, which screws into a companion thread cut in the casing. The patterns shown in figs. 8 and 9 are practically absolete.

Another form of water regulation which is sometimes used is obtained by constructing the combining and diverging cones similar to those in Mr. Webb's injector, fig. 6, and moving them axially by a rack and pinion situated in the overflow chamber. This was first done by Mr. Gresham.

A modified form of the original "Giffard," made by Messrs. Schäffer and Budenberg, is shown in fig. 10. The combining and diverging cones are screwed together, just below the overflow, and then inserted into the outside casing, being held there by the guide frame of the back-pressure valve. The outer casing is made in two parts, bolted together at the top of the steam cone. This fulfils two objects: the steam flange may be turned round at any time to suit the conditions of circumstances, without in any way impairing the action or usefulness of the apparatus; and if the top casting is removed the remainder forms a complete non-lifting injector. Of course, it is not necessary to metamorphose an injector in this way when once completed, but the same patterns

and templates would do for the lifting as well as for the non-lifting types. The steam regulating spindle is tapered at its lower extremity for the purpose of graduating the admission of steam while the feed water is being lifted

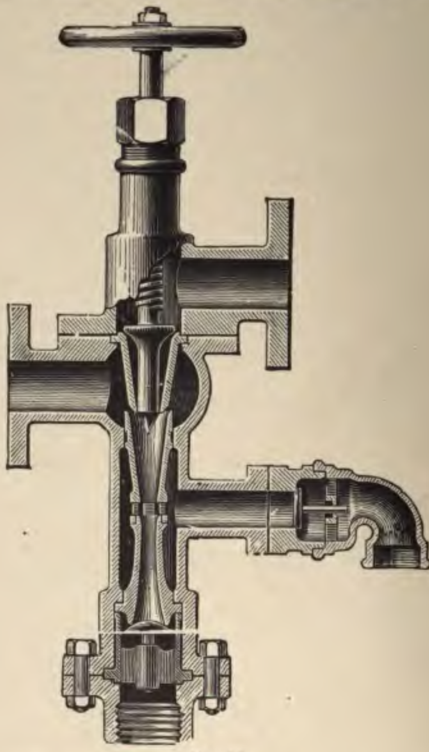


FIG. 10.

to the combining cone at starting; and it also has another conical portion, which fits down upon the top of the steam cone, thus forming a stop valve for the steam supply. With this arrangement the steam valve on the boiler may be dispensed with. A non-return valve is also placed in

the overflow pipe, preventing any air being taken into the boiler with the feed water. The overflow valve also augments the efficiency of the injector as a pump to a small extent.

Another method of exhausting the water pipe before the injector starts is shown in fig. 11. The steam-regulating spindle has a small hole drilled for some distance along its axis at its conical end, terminating near its parallel part in two transverse holes. When it is desired to start the injector, the spindle is slightly withdrawn, so that steam may enter by the transverse

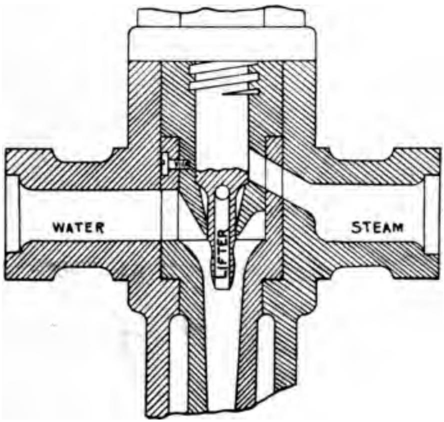


FIG. 11.

holes, and issue into the combining cone by the longitudinal channel. The orifice of this channel being much smaller than that of the overflow, a vacuum is produced in the water pipe, as previously explained. On the arrival of the water at the overflow the spindle is withdrawn, so that the full amount of steam can enter the injector. The arrangement shown in the figure is now in common use by a number of makers.

In 1880 Messrs. Davies, Hamer, and Metcalf were engaged in perfecting an exhaust steam injector, in which

the combining cone was split into two halves for a portio of its length by a plane passing through its axis ; the piec which was thus split off being replaced and hinged to th main portion of the cone at the upper end of the split Experiment showed this improvement to be so automati in its action that it was attached to the ordinary pattern of non-lifting injectors, thereby converting them int *lifting automatic re-starting* injectors, which, after ver slight modification, now appear as shown in fig. 12.

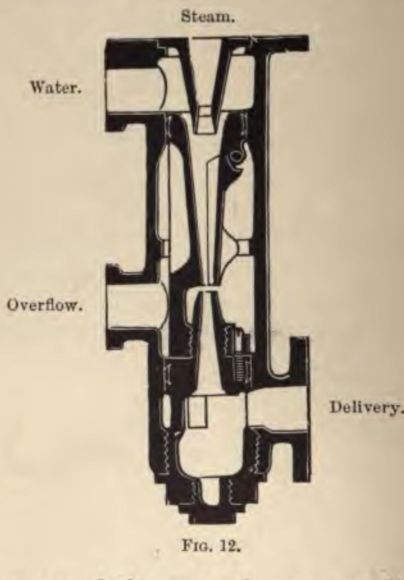


FIG. 12.

The cones are of the same shape or profile as in th non-lifting pattern, but the method of securing th delivery and combining cones is somewhat different, th two being screwed together just below the overflow gap The whole is then inserted into the outer casing, an secured to it by a few threads at the delivery end. Th other joints are rendered tight by inserting packing in th double V grooves. The combining cone is cast in tw

pieces, the two being hinged together as shown in fig. 12. Perhaps a better idea may be gained of it from the elevation of it in fig. 13, at E. The two pieces are machined and fitted together and then bored, so that the cone shall be perfect when the flap is closed. When steam

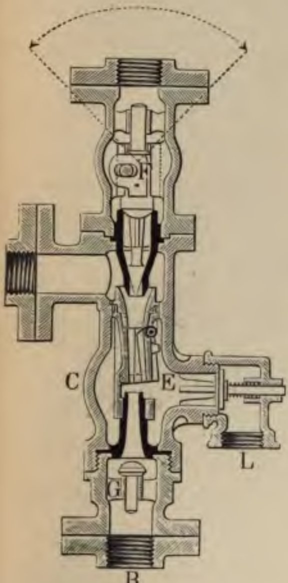


FIG. 13.

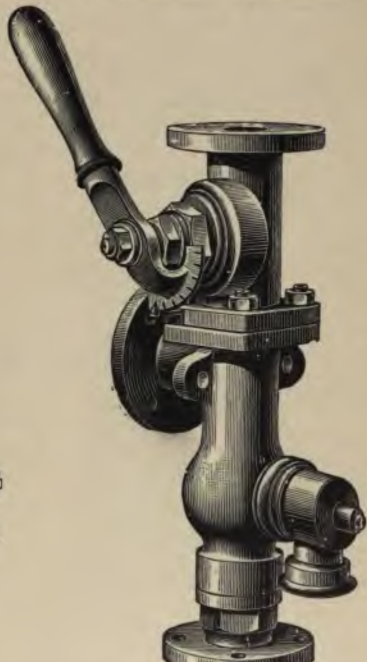


FIG. 14.

Messrs. Schäffer and Budenberg's "Perfect" Injector.

is turned on after issuing from the first cone, it enters the second and pushes up the flap, thereby securing a large area of outlet—much larger than that of the steam cone (an absolute necessity with full steam on)—which enables the water pipe to be exhausted, and the feed water raised

to condense the outflowing steam. On the arrival of the cold feed water the steam is instantly condensed; at the same time the pressure in the overflow chamber closes the flap, and retains it closed until, by some cause or other, the water stream is broken and the injector stops. The flap is then forced open, the feed drawn up, and the injector is at work again. The attendant working the injector always becomes cognisant of the injector starting to force water into the boiler by the distinctly audible clack as the flap closes up. These movements of the flap are shown in figs. 15 and 16, which show in detail the

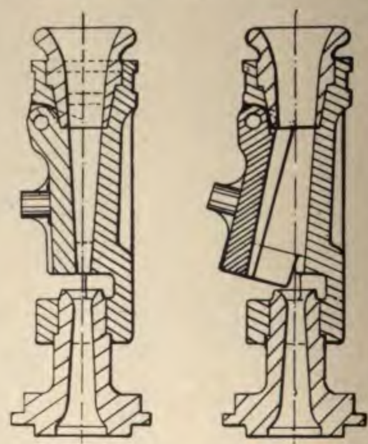


Fig. 15.—Flap closed. Fig. 16.—Flap open.

diverging and improved combining cone of Messrs. Schäffer and Budenberg's injector. The flap is shown closed, exactly as it is when the injector is at work; while fig. 16 shows the flap open, as it would be when the injector is idle and placed in a vertical position. On the end of the flap, remote from the hinge, are cast a pair of ears, which act as guides should the hinge become at all worn. The uppe portion of the combining cone is let into the main portion so as to secure the continuity of the jet. The author ha

had the opportunity of examining some of these flap nozzles minutely, and he feels bound to express his admiration for the very perfect workmanship exhibited throughout in their construction. The flap fits upon the main part so well that the joint is hardly perceptible when closed, and quite air-tight.

This instrument is specially well adapted for locomotives on account of its re-starting capabilities. A driver always has enough to do to look after the regulator, the reversing lever, and signals, and the fireman finds that on the average express journey he has none too much time on his hands in which to be watching and manipulating the injector; so that a really automatic re-starting injector is often a great boon.

Another important feature in this injector is the facility with which the two water cones can be withdrawn for cleaning without breaking any pipe joints. In case the combining cone should stick and become unscrewed from the diverging cone, a little screw is inserted at the joint, so that they must both come out together. On account of the automaticity of action, virtually no water is wasted at the overflow at any time.

Messrs. Schäffer and Budenberg use the flap nozzle in their "Perfect" re-starting injectors. A section and a perspective elevation of their injector are given in fig. 14. As in the same firm's modified form of the original "Giffard," the lower part of the casting contains the whole of the cones—in fact, the injector proper; while the top casting is added for the sake of the regulating device, and can be turned round into four different positions, so that the same injector is at once a *right* and *left* hand pattern. The steam from the boiler passes through the top casting containing the regulating spindle F, also carries the steam stop valve. This spindle, which increases the general efficiency of the apparatus, especially when working under abnormal conditions, is actuated by the hand lever which turns the little eccentric pin shown in the figure, the pin moving in a slot. The pin is so situated that it effectually locks the steam valve when closed. In the perspective view, fig. 13, there will be noticed, attached to the handle,

graduated arc, which registers the correct position of the lever for different steam pressures by means of a fixed pointer; so that in starting the injector the handle has to be moved over such that the index points to the ... boiler pressure, and the injector easily starts. This will do without any water regulation between the pressures of 30 lb. and 165 lb.; below this limit a little water regulation may be necessary. The regulation by means of a handle in preference to the screw was adopted on account of the simplicity of manipulation with the ...

... return valve is generally used on the overflow ... and kept on its seating by a weak spring, the valve ... passing through the casing, so that it may be made ... again if by sediment or other means it should become ... In the locomotive pattern there are more overflow ... than one, the remainder being covered with blank ... This allows of the body of the apparatus being ... into any position when necessary. The casing is ... at the overflow so as to act as an air space ... flap, which assists in securing good working. ... the water in the tender of a locomotive may be ... surplus steam, a screw-down overflow valve is ... of the pair of injectors. The non-return ... besides increasing the efficiency under ... tances, causes it to work dry under certain ... s, such as high temperature feed water. ... of water at re-starting, as it closes ... nsation begins; and the momentary ... ets in the overflow chamber, and

diverging ... and Bude ... as it is w... the flap op... placed in a ... remote from ... as guides sho... portion of the ... so as to secure ...

what injectors of this type ... ll open. With steam at ... ft 18 ft. with feed water ... the same distance at ... at a temperature ... pressure ranging ... 140 deg. Fah., ... 3 ft. without ... nd steam at

180 lb., it will take feed at the high temperature of 105 deg. Fah. The overflow valve also assists the injector to feed at higher temperatures. The injector starts or re-starts perfectly and absolutely without loss of water.

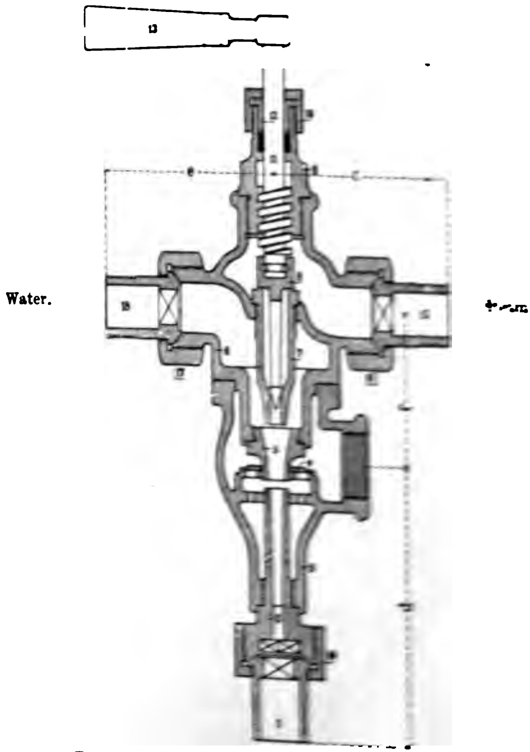


FIG. 17.—Schäffer and Budenberg's [illegible]

It can be placed either vertically or horizontally [illegible] in the latter position it must [illegible] underneath.

Messrs. Schäffer and Budenberg make [illegible] injector, of which a section is shown in fig. [illegible]

a graduated arc, which registers the correct position of the lever for different steam pressures by means of a fixed pointer; so that in starting the injector the handle has only to be moved over such that the index points to the then boiler pressure, and the injector easily starts. This it will do without any water regulation between the pressures of 30 lb. and 165 lb.; below this limit a little water regulation may be necessary. The regulation by means of a handle in preference to the screw was adopted on account of the simplicity of manipulation with the former.

A non-return valve is generally used on the overflow orifice, and kept on its seating by a weak spring, the valve stem passing through the casing, so that it may be made to fit tight if by sediment or other means it should become tiled. On the locomotive pattern there are more overflow orifices than one, the remainder being covered with blank flanges. This allows of the body of the apparatus being turned into any position when necessary. The casing is bellied out at the overflow so as to act as an air space round the flap, which assists in securing good working. So that the water in the tender of a locomotive may be warmed by surplus steam, a screw-down overflow valve is placed on one of the pair of injectors. The non-return overflow valve, besides increasing the efficiency under ordinary circumstances, causes it to work dry under certain extreme conditions, such as high temperature feed water. It also prevents loss of water at re-starting, as it closes at the instant condensation begins; and the momentary discharge of water collects in the overflow chamber, and is drawn into the boiler.

The following will indicate what injectors of this type will do with the water valve full open. With steam at 60 lb. per square inch, it will lift 18 ft. with feed water not exceeding 105 deg. Fah., or lift the same distance at 180 lb. per square inch with feed water at a temperature not exceeding 80 deg. Fah. With a pressure ranging between 30 lb. and 60 lb., and feed water at 140 deg. Fah., the injector will still lift a height of 3 ft. without regulation of feed water. With no lift, and steam at

180 lb., it will take feed at the high temperature of 105 deg. Fah. The overflow valve also assists the injector to feed at higher temperatures. The injector starts or re-starts perfectly and absolutely without loss of water.

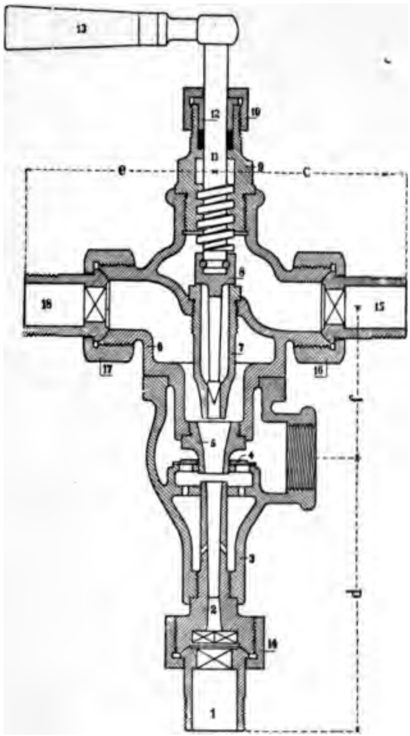


Fig. 17.—Schäffer and Budenberg's "Tandem" Injector.

It can be placed either vertically or horizontally, but when in the latter position it must never have the flap underneath.

Messrs. Schäffer and Budenberg make another type of injector, of which a section is shown in fig. 17.

It differs from those previously described in the manner in which it provides a large outlet for the steam while it is sucking up the feed, and before beginning to actually feed the boiler. The combining cone has a transverse slice removed, through which the steam may readily escape to the overflow pipe. This aperture in the combining cone is surrounded by a chamber open to the ordinary overflow orifices, but closed to the atmosphere by a small disc valve. A valve of some description is necessary to prevent the atmosphere from flowing into the partial vacuum in the combining cone while the injector is at work.

This injector can be supplied without the steam-regulating handle and spindle, but the instrument does not permit of so much adjustment in capacity without this handle.

We next come to an injector which is automatic in re-starting, and at the same time contains in itself both water and steam regulator, operated by a single handle. A longitudinal section of this injector, which is the invention of Mr. R. G. Brooke, of the firm of Messrs. Holden and Brooke Limited, is shown in fig. 18. Its capacity for re-starting automatically is derived from the transversely divided combining cone, the outlet to the little chamber surrounding this division being closed by a flap valve, while the ordinary overflow orifice is in communication with the atmosphere, as usual.

The action of the flap valve is precisely similar to that of the flap nozzle. Of course, an ordinary disc valve may be used, as in the previous case. The makers have made the seating of the flap so that it may be turned round into any position at a moment's notice if necessary, thus allowing the valve to be so placed that it will always fall upon its seat, this being its most efficient position. The exterior surface of the steam cone is cylindrical, and slides axially, without turning, in the stuffing box and gland, as shown in the figure. The upper portion of this cone has apertures through which steam may pass to the central channel, the opening to which is regulated by the coned spindle, which also carries the stop valve. In the prolongation of the upper part of the cone are cut some

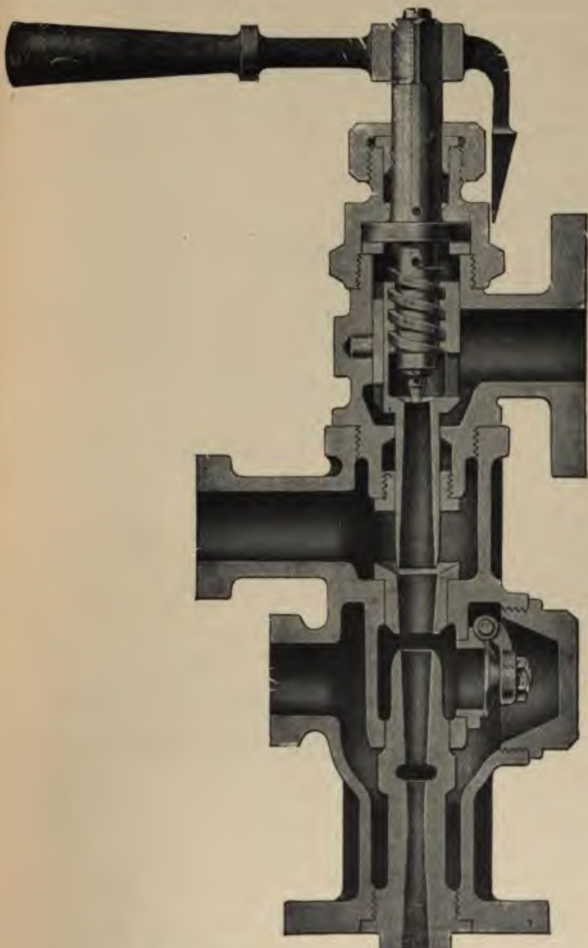


FIG. 18.—Messrs. Holden and Brooke's "Sirius" Injector.

threads of a quick pitch, into which gear corresponding threads cut on the spindle, the cone being prevented from turning round by the feather key fixed in the outer casing.

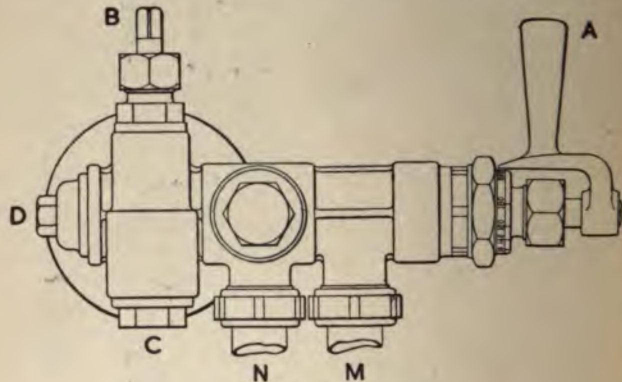


FIG. 19.

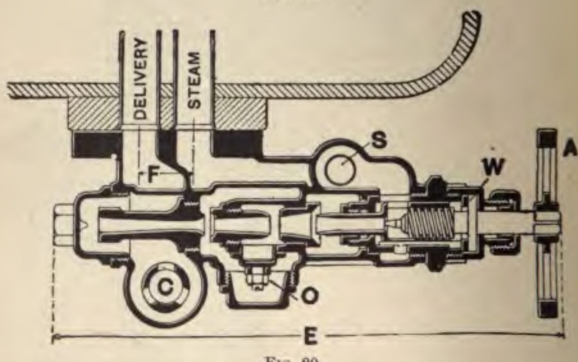


FIG. 20.
"Sirius" Combination Injector, Horizontal Pattern.

A collar on the spindle prevents its axial motion. The threads being of a large pitch, only a partial turn of the handle is required to render the passages full open. Analysis shows that with different pressures the steam

supply should vary in the opposite manner to the water supply. This is admirably brought about by a partial turn of the steam spindle, fig. 18. The lever or handle is prolonged beyond its boss, tapered, and bent over, the extremity pointing to a series of numbers on the cylindrical part of the nut, which indicate the pressure of steam for that particular position of the handle pointer. By this method of regulation the injector works equally well with almost any variation of steam pressure. It will thus be seen that by a single movement of the handle the steam is turned on, the injector started, regulated for any steam

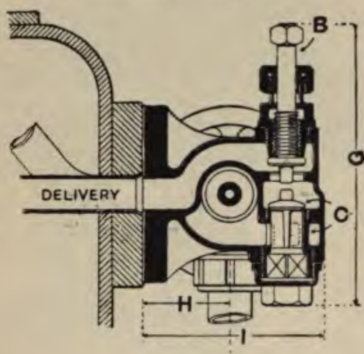


FIG. 21.

pressure, and the quantity and temperature of feed water (as delivered to the boiler) varied between the two extremes of maximum and minimum. The moving parts are situated in the steam space, so they are not liable to become deranged through deposit from dirty water.

The makers of the injector just described have adapted it to the horizontal combination form shown in figs. 19, 20, and 21. It is fixed on the side of the firebox, as shown in fig. 20. Figs. 19 and 20 are not exactly alike in all details, but the dissimilarity is unnoticeable. In all these injectors stop valves on the steam and delivery passages are provided, which enable the cones to be withdrawn under steam.

In cases where space is restricted, the same firm supply the very neat form of horizontal pattern, shown in figs. 22 to 24. The screw down valves E and B are for the purpose of isolating the injector for the removal of cones under steam or grinding in of check valve D. Steam is regulated by the valve A and the feed supply to the injector by the cock F. The dotted fitting L is for the purpose of connecting a fire hose. This is very general

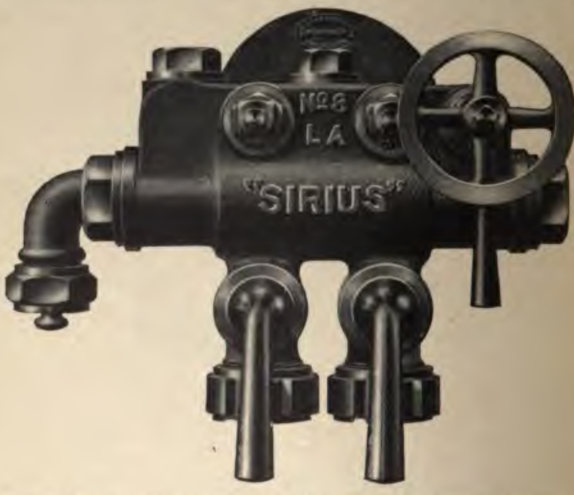
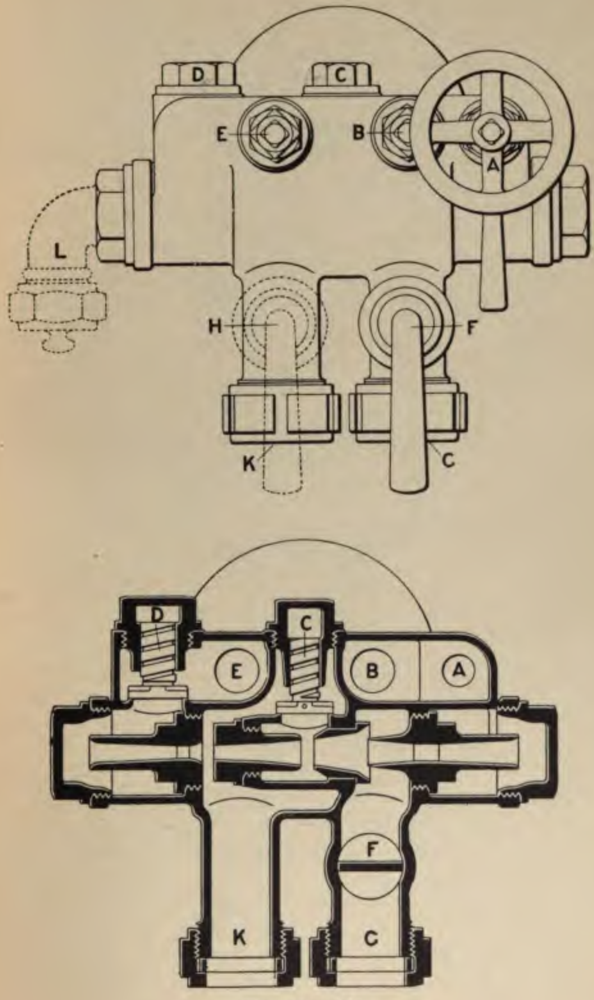


FIG. 22.—Messrs. Holden and Brooke's Horizontal Combination Injector for restricted space.

on some of the Continental railways. The feed warming cock H is shown dotted in the elevation, and is only supplied when specially ordered.

A cheaper and smaller form of the same injector is shown in elevation and plan in fig. 26, and is much used on stationary boilers. In the plan H is the delivery check valve and I the delivery screw-down valve, F the steam valve, and G the water cock. Steam enters the injector by the passage E (plan), and the companion passage serves to deliver water to the boiler.



FIGS. 23 and 24.

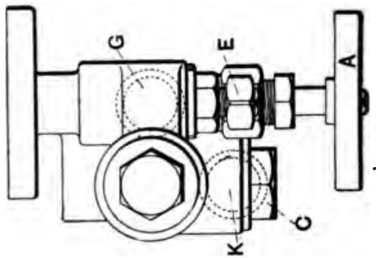


FIG. 25.—End elevation of Horizontal Combination Injector (steam end).

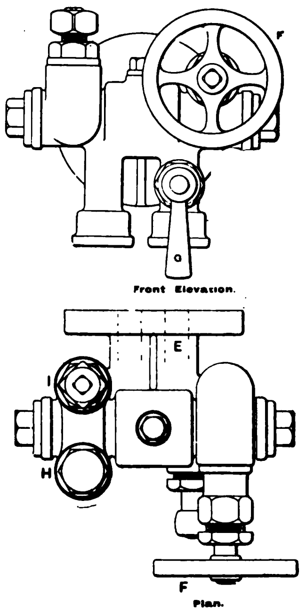


FIG. 26.

The same firm also make a vertical type of combination injector, fig. 27, which is generally situated on the back plate of the boiler in the cab, though sometimes on the side of the firebox.

A new combined steam and water regulator has recently been introduced by Messrs. Holden and Brooke. It is shown in fig. 28.

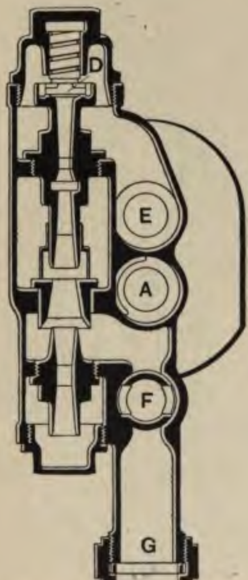


FIG. 27.—Messrs. Holden and Brooke's Vertical Combination Injector.

The steam regulating spindle A supports a yoke P. As the spindle is withdrawn by screwing it backwards, the yoke P is lifted, and the water valve B is raised, thus admitting water to the injector.

It is shown later that when working at low pressures a larger steam area is required than for high pressures. In the arrangement under consideration the steam and

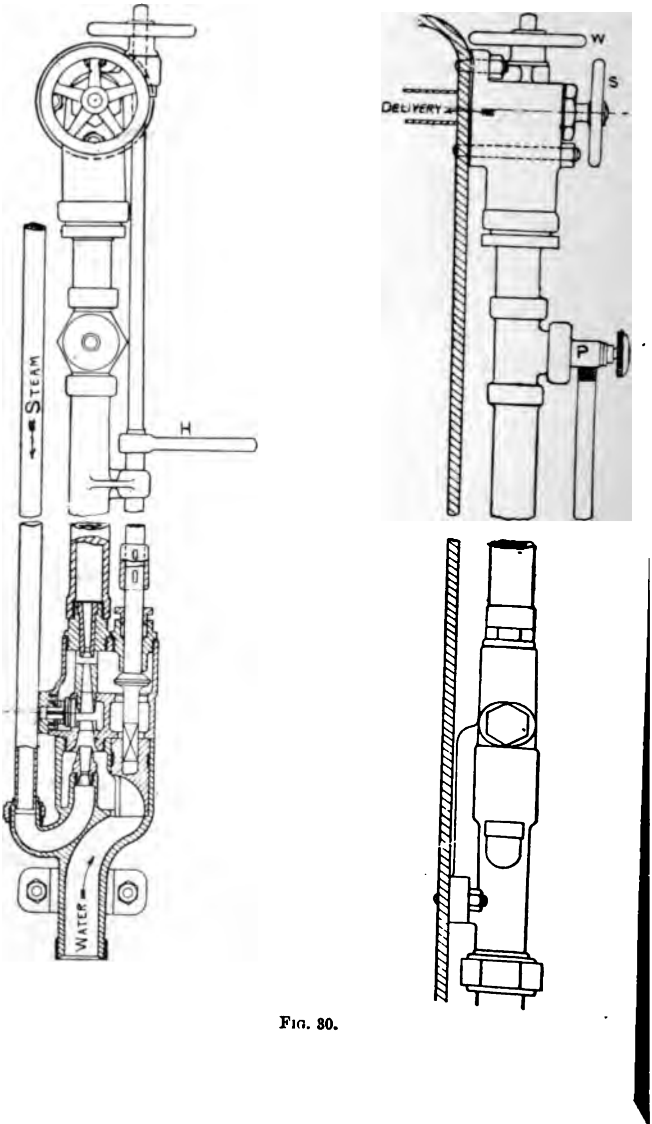


FIG. 30.

Mr. F. W. Webb replaced most of the injectors on the London and North-Western Railway (fig. 10) by the one shown in fig. 30. The handle H operates the overflow cock and the hand wheel to the overflow valve. This is almost identical in the main parts with that described above (fig. 10).

Another combination injector, and one which is used on English railways to a greater extent at the present time than any other *combination* injector, is that of Messrs. Gresham and Craven, two longitudinal sections of which are given in figs. 31 and 32. In this form it has remained, with slight modification, since it was brought out in 1884. Steam is admitted through the lower channel in the flange by the steam valve H, from which it passes to the steam cone C, through the channel S. The non-return screw-down valve Y covers the delivery orifice, in case the clack valve (in the left-hand section) should become tiled or the cones require to be removed under steam. The latter can be accomplished by first taking off the nut just above the cone R, and then unscrewing each cone in turn from the main casting. There are hand valves, W and P, on the feed-water and overflow pipes respectively; the former for regulating the supply of water; the latter for heating the feed water with surplus steam. The principal feature of this apparatus is the device from which it derives its automatic re-starting property.

In the lower portion of the diverging cone is the small loose cone A, capable of sliding axially, its axis being maintained coincident with that of the other cones by four wing guides, three of which are shown in fig. 32. In the figure, the loose cone is shown in the position it occupies when the injector is exhausting the water pipe, the steam being able to escape into the overflow channel O, through the space between the combining cone B and the loose cone A, and also at the end of the latter, the wings preventing the flange of the cone A from touching the diverging cone. Directly the water arrives and condenses the steam, the partial vacuum formed in the combining cone allows the loose cone to fall until its flange rests upon

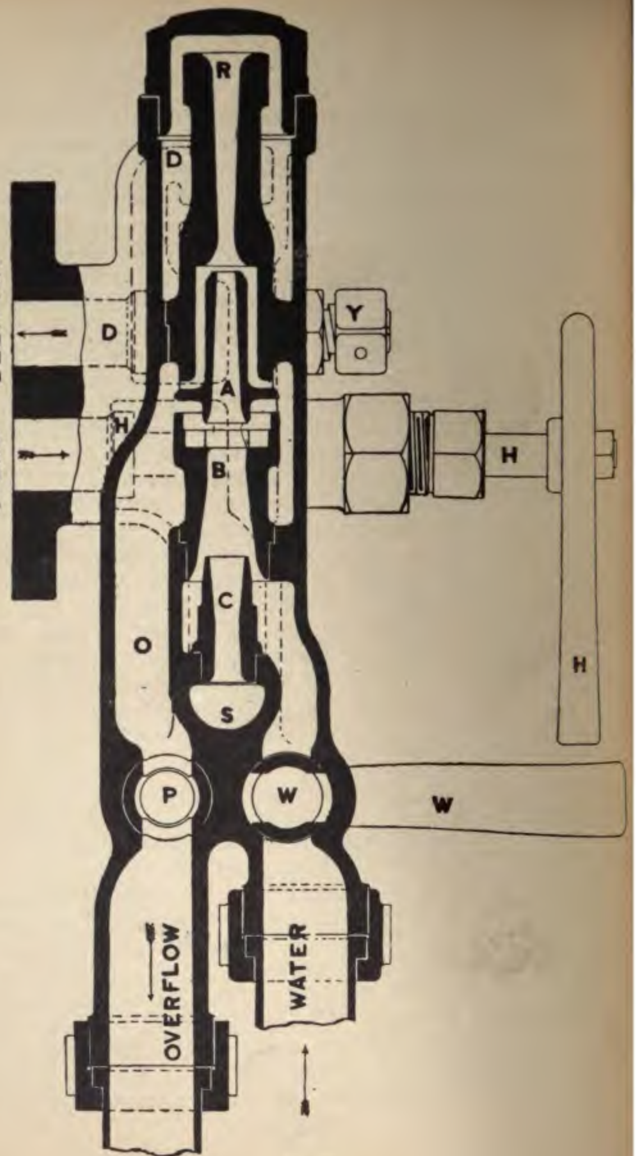


Fig. 31.

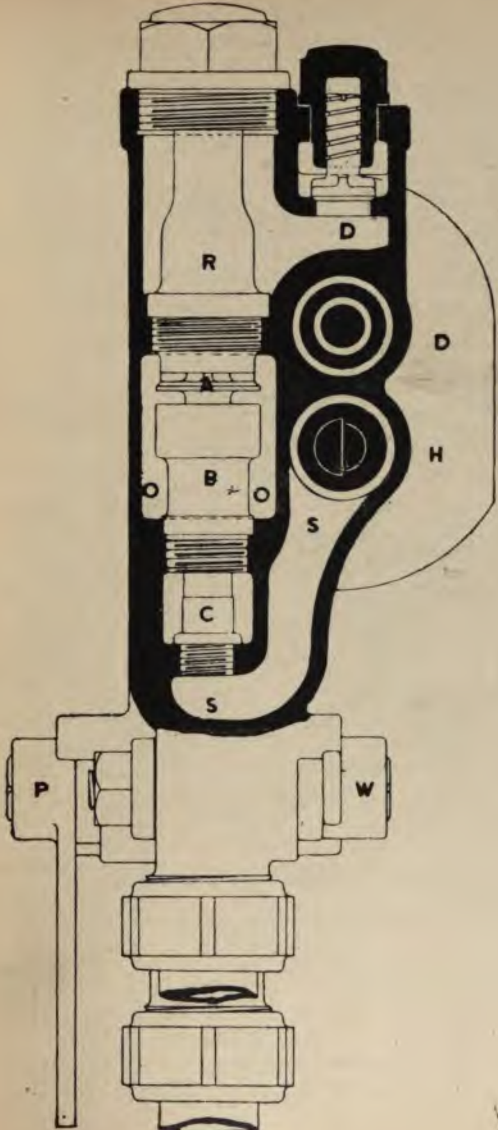


FIG. 32.

the combining cone, it being assisted and held there by the pressure of the atmosphere from without. In this closed position the two cones A and B together form one long continuous cone.

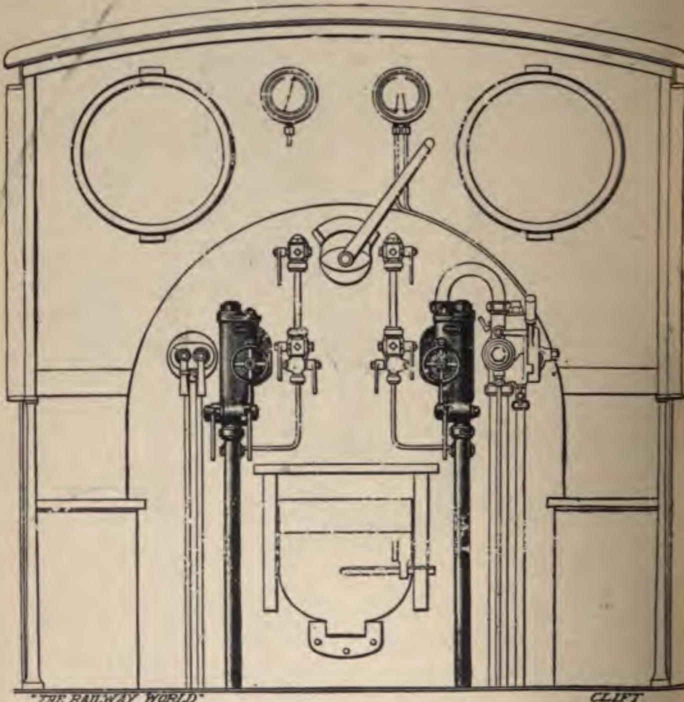


FIG. 33.—Messrs. Gresham and Craven's Vertical Combination Injector in position.

This injector works quite dry—a very desirable feature—and is very much liked by drivers and firemen. It is with this object in view that it has been placed on many engines running trains through suburban districts of large towns, the continual blowing through of steam when

good water regulator is added to this pattern, and is fitted with a dial to indicate the opening of the feed channel.

A smaller pattern of Messrs. Gresham and Craven's horizontal type is shown in fig. 35, and the general arrangement in fig. 36. The illustrations are practically

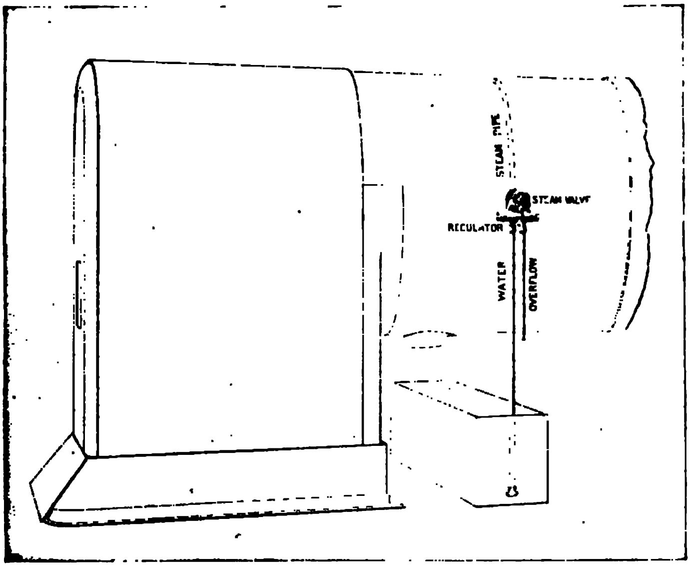


FIG. 36.

self-explanatory. This pattern is much used on stationary boilers.

An injector manufactured in America is the "Metropolitan," the single tube variety being shown in section in fig. 37. The first aperture in the combining cone is closed by the disc valve R while at work, but there are two other smaller openings in the same cone for the

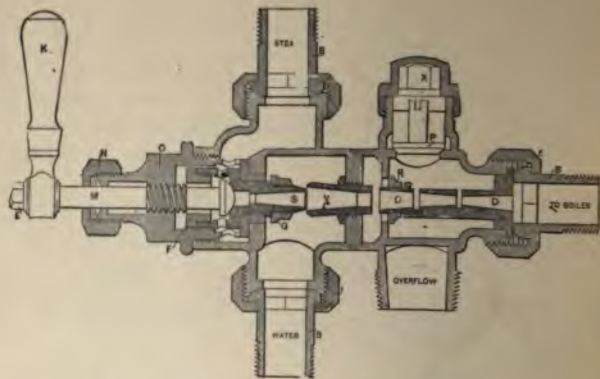


Fig. 37.—"Metropolitan" Single Tube Injector.

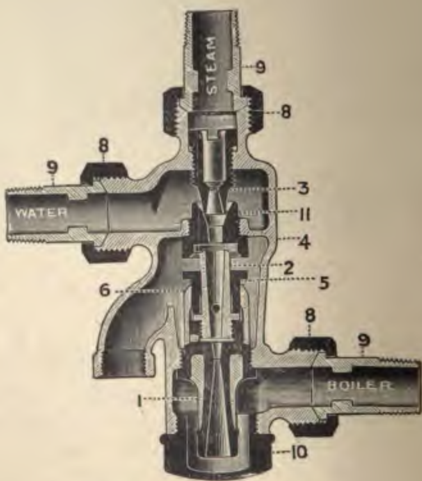


Fig. 38.—The Sellers Injector.

purpose of facilitating its starting. There is also an overflow valve at P. The steam cone is capable of a certain amount of axial motion. It is actuated by means of the steam valve and handle K, and permits of a greater or less aperture between the steam cone and the

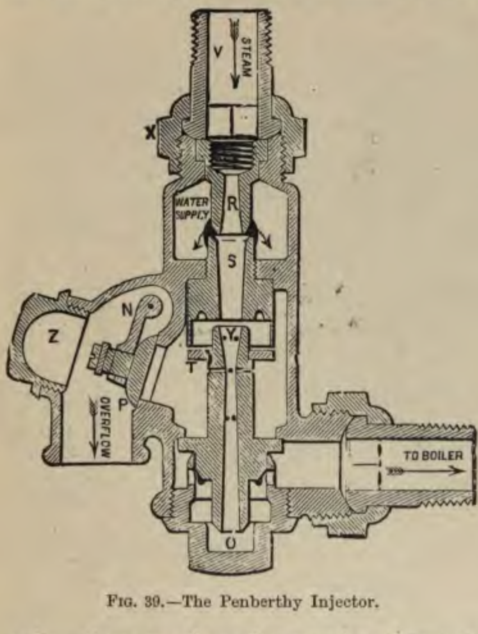


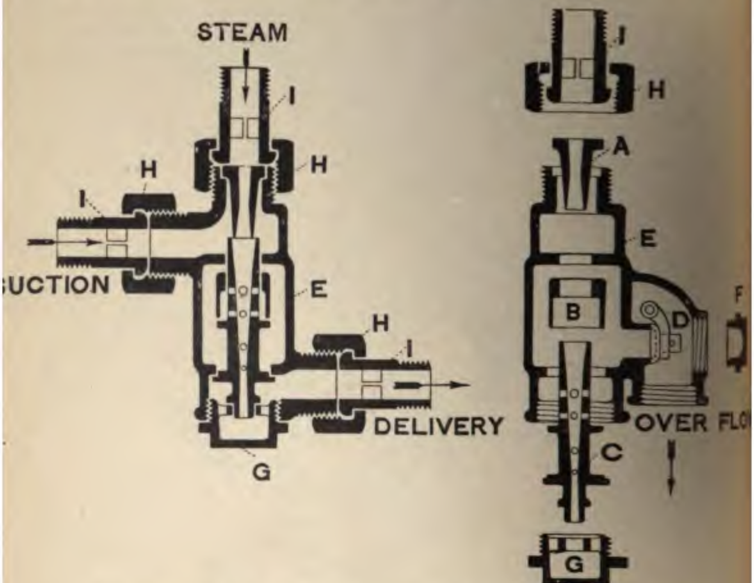
FIG. 39.—The Penberthy Injector.

suction tube, thus varying the supply of water to the latter. The nut G prevents too great a movement of the steam cone.

Another injector not unlike the last illustration is the "Sellers," shown in fig. 38, and sold in this country by Messrs. T. A. Ashton, of Sheffield. The starting overflow is covered, while at work, by the sliding sleeve valve 5, the rush of air back along the overflow pipe, directly the steam is condensed, impinges on the flange of the sleeve 5, and carries it up against the lifting tube 11 and holds it there.

The outer sleeve 6 is also capable of movement to permit of overflow taking place from the lower aperture, but while at work it is held down in the position shown by the atmospheric pressure.

The "Penberthy" injector (sold by Messrs. W. H. Willcox, London) is given in section in fig. 39, and it will



Sections showing how the cones are removed by withdrawing the end nuts.

Fig. 40.—Messrs. Holden and Brooke's "Premier" Injector.

be seen to be very similar in construction to those just described in figs. 37, 38, 40, and 17.

A fourth injector of the same character is shown in fig. 40, the right half of the figure representing the different parts somewhat separated to show how they are held in position by means of two nuts only. The figure is so clear that any further explanation is superfluous.

A type of instrument which provides a large overflow orifice is the "International," shown in fig. 40*a* (Messrs. Armstrong and Co., London).

In addition to the overflow valve, the delivery chamber is connected by a throat to a supplementary chamber

Delivery. Steam.

Overflow. Suction.

FIG. 40A.

having a valve in line with the overflow valve, so that at starting the overflow valve in opening drives the second valve off its seat, and thus provides an outlet for the steam that passes into the delivery chamber through the combining cone. As soon as the cold feed reaches the combining tube, the partial vacuum formed closes the

normal overflow valve, and the pressure of the delivery quickly closes the supplementary overflow connected to the delivery chamber.

CHAPTER III.

Construction and Arrangement of High-pressure Compound Injectors.

In the simple form of injector described in the last chapter, the pressure at the beginning of the delivery cone —that is, at the last overflow orifice—was that of the atmosphere if the chamber were open to the atmosphere; but if there was an overflow valve which excluded the atmosphere, the pressure in the overflow orifice between the cones would practically be that due to the vapour of water at the temperature of the jet.

In either case the atmospheric pressure would be about the maximum in that neighbourhood under ordinary conditions, together with the corresponding temperature of 212 deg. Fah.

If the overflow valve is loaded, a greater pressure than that of the atmosphere may be produced in the overflow chamber, and the temperature of the feed entering the boiler would then be higher than 212 deg. Fah.—a very desirable result. But if the overflow valve be loaded, it will not open freely at starting, and the apparatus will not commence to work.

This difficulty has been overcome by using two injectors in series, the first having an overflow orifice, and termed the "lifter"; while the second has either a loaded valve or no orifice at all, and termed the "forcer."

The lifter delivers water to the forcer under a pressure generally varying between 20 and 40 lb. per square inch. The final temperature of the water delivered to the boiler may be as high as 280 deg. Fah. In addition to the advantage of high temperature feed, the pressure of the water delivered to the forcer will assist in increasing the pressure of the delivery; that is, a compound injector

may be made to feed against very high pressures with greater ease than a simple injector.

The ideal form of the compound injector is shown in fig. 41, and the actual form may be like this or as shown in fig. 42.

FIG. 41.—Ideal form of Compound Injector.

A longitudinal section of Messrs. Körting's injector is shown in fig. 43, and the general arrangement in fig. 44. The left line of tubes forms the lifter and the right set the forcer. It will be noticed that the steam cone of the

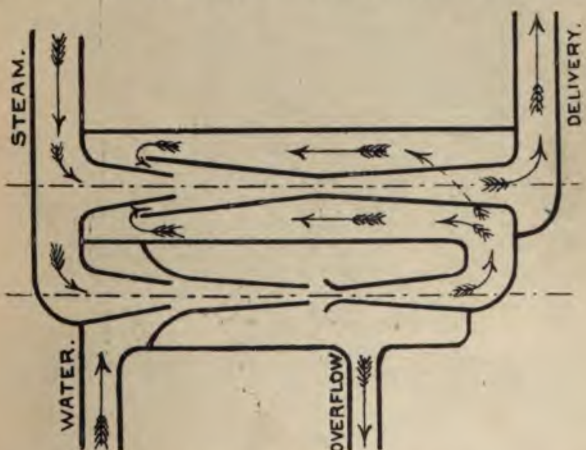


FIG. 42.—Actual form of Compound Injector.

lifter is much smaller than that of the forcer, on account of the lifter having to suck up its feed without any overflow openings in the combining cone. There is an

overflow cock E which only comes into action at starting. The two steam valves are operated by a cross piece, connected at its centre with a spindle operated by the starting handle, simultaneously with the overflow cock E. The construction of the operating gear can best be seen

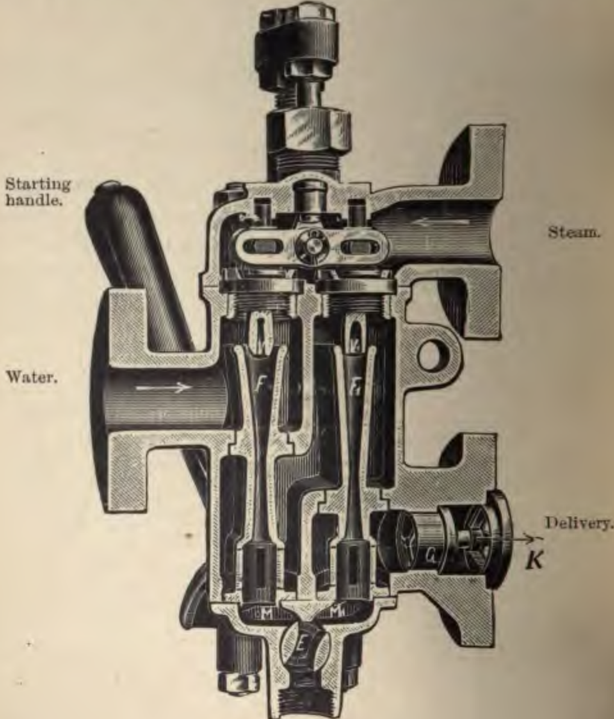


Fig. 43. – Messrs. Körting's Compound Injector.

in fig. 44. The handle B has an eccentric on its spin[dle] which operates the steam valves through the eccentric shown in the figure.

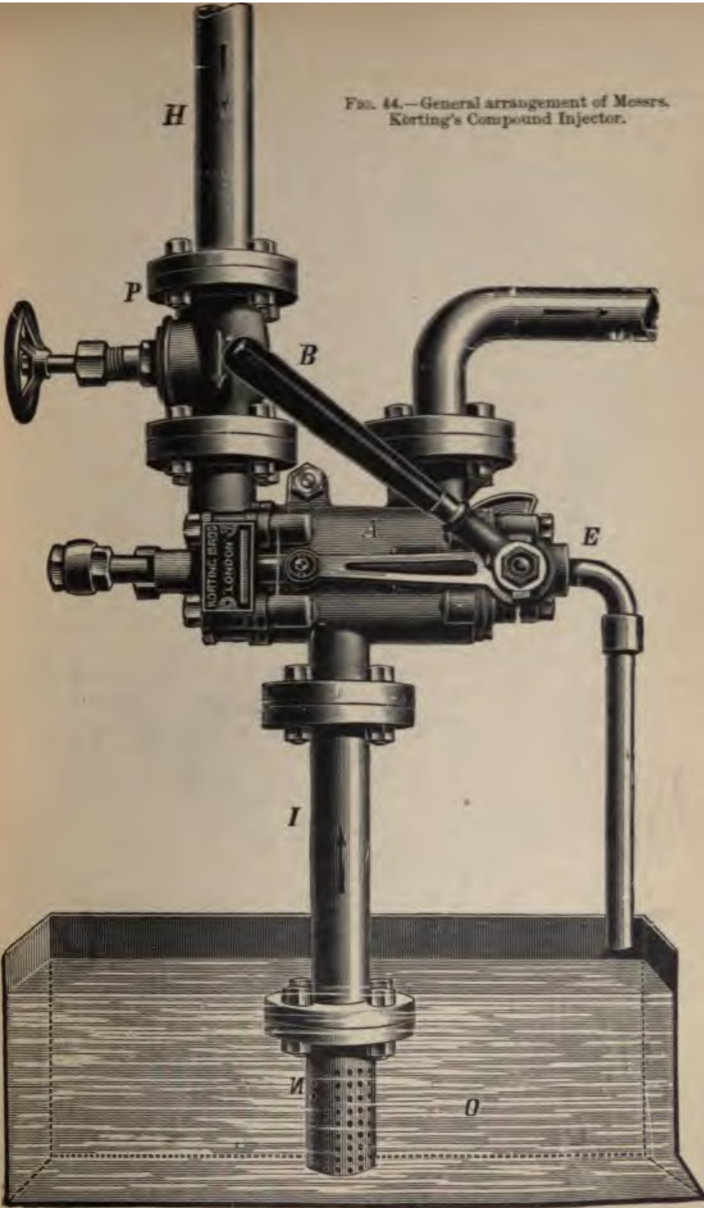
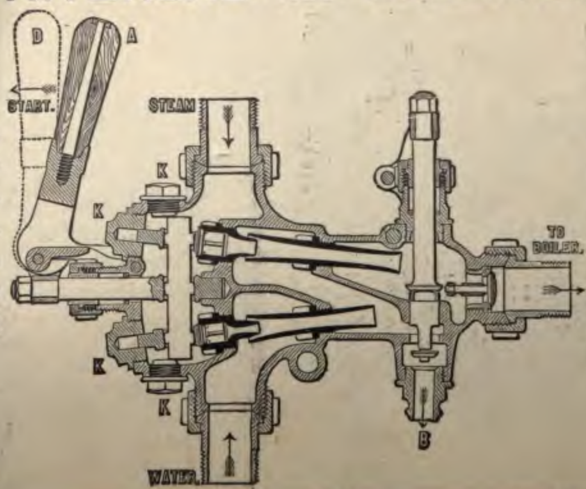


Fig. 44.—General arrangement of Messrs. Korting's Compound Injector.

UNIVERSAL DOUBLE TUBE INJECTOR.

When removing Steam and Water Nozzles for repair or cleaning, please note that the lower nozzles are marked **L**, and the upper ones **U** (stamped on casting). They should be thus replaced.



During working hours the Stop Valve on boiler (which may be any kind of a valve) remains turned on, and the stopping and starting is solely effected by the lever, **A**, operating the valves in the steam-chamber of Injector.

Fig. 45.—Schutte's Körting Injector.

Returning to fig. 43. At starting, the handle is moved over slightly, and as one steam valve is larger than the other, the smaller one will open first and allow steam to pass to the lifter to suck up the feed, the steam and water passing out through E to the atmosphere. When water is observed to flow, the handle is put right over, thus opening the steam valve of the forcer. The final movement of the handle closes the overflow E, and the jet is obliged to enter the boiler.

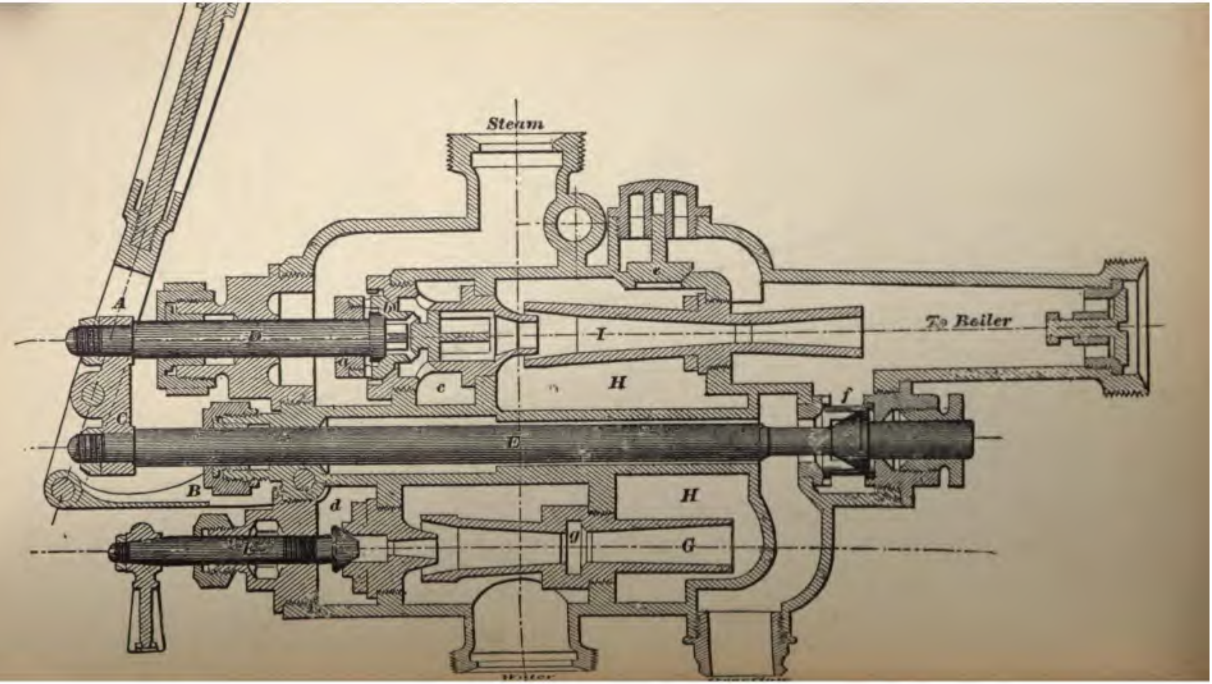
A pattern of the Körting injector, made in America by Messrs. Schutte, is shown in fig. 45. The drawing explains itself. These injectors may be placed in any convenient position, and will lift over 20 ft. with cold feed water. The maximum temperature of the feed water with an overhead supply is 150 deg. Fah.

It is not necessary to regulate the steam or water supply for variation of steam pressure, as the quantity passing through the injector depends upon the lifter; so that if by a decrease of pressure less water enters the lifter, there will be less for the forcer to drive into the boiler.

Some experiments on one of these injectors show that the amount of water delivered per hour varies according to the temperature of the feed. A non-lifting injector, No. 4, delivered 2,000 lb. of cold water, but only about 1,600 lb. of hot water per hour with steam at 60 lb. per square inch.

The same makers use these injectors in conjunction with feed-water heaters on the German State railways. The heaters are supplied with exhaust steam from the blast pipe.

A very good improved form of compound lifting injector is shown in section, fig. 46. The chief features of this apparatus are the automatic action of the overflow valve *e*, and the entire regulation of steam to both lifter and forcer by the single valve stem D. The little supplementary regulator F is only attached to locomotive injectors. There is also an attempt to automatically regulate the supply of feed water delivered to the forcer by the lifter with the orifice *g* between the combining and delivery cones of the lifter. Should too much feed pass


Steam
To Boiler
A
B
C
D
E
H
H
I
G
c
d
e
f
g

with the steam into the combining cone, the surplus will return to the feed chamber through the passage *g;* but should there not be sufficient feed entering with the steam, then more will be induced into the cone G through the aperture *g*.

To start the injector, the handle is slightly lifted, thereby raising the collar valve on the end of the stem D from its seating, and allowing a small amount of steam to pass through the main valve *b* into the chamber *c*, and thence into *d*, from which it flows into the lifter and raises the water to the injector. On the arrival of the water, which pushes back the overflow valve *e*, thence through *f* and out at the overflow pipe, the handle is slowly raised to its extreme position, opening the main steam valve *b* and gradually closing the overflow valve *f*, after which the injector is at work. The above injector is of American design, the makers being Messrs. Hayden and Derby, of New York.

The injector shown in section, fig. 47, and elevation, fig. 42, is the "Buffalo" injector—of American origin, but sold in this country by Messrs. Green and Boulding, of London. The design is very compact and handy, and all the moving parts can easily be examined whilst the injector is under steam. Steam is admitted at the lifter side on opening very slightly the operating lever, and to the forcer side by the continued movement of the lever to the stop. The overflow valve controlling the overflow orifice of the forcer is operated by the spindle 17, which allows it to close automatically when the machine is at work, and opens it when the machine is shut off. When the water has arrived, and condenses the steam, a pressure is produced in the discharge chamber of the forcer, which closes the overflow valve.

The "Albion" compound injector arrangement is shown in fig. 48. Here the lifter is separated from the forcer, and situated in the well or tank from which the feed is taken. This has one special advantage, namely, that it enables very hot water to be used from a level much below the forcer. This cannot be accomplished with a simple injector unless the injector be situated in the position of the lifter in fig. 48, and this is not desirable.

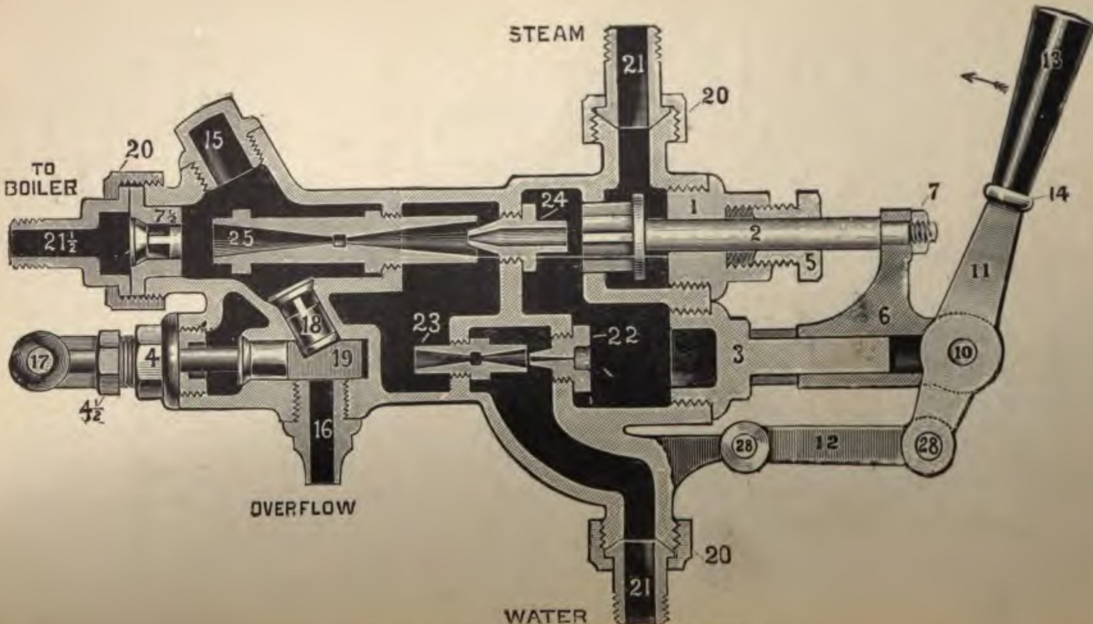


Fig. 47.—The "Buffalo" Compound Injector.

he lifter in fig. 49 is quite simple in construction, and steam pipe has a protecting sleeve where it is ıersed in the feed tank, for the purpose of surrounding ith a layer of air, and thus prevent the steam from lensing before it reaches the lifter.

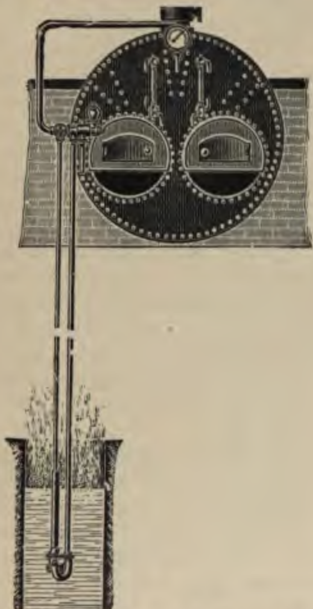


FIG. 48.—Compound Injector made by the Albion Engineering Co., of Manchester.

he forcer is a carefully designed instrument aining a steam valve, which is arranged to supply ı lifter and forcer. The valve is hollow and pierced ı holes round its circumference. The end of the dle prevents any steam from passing to the lifter, the small projection on the valve covers up the steam of the forcer and so prevents steam reaching it even reached the lifter.

A partial turn of the starting handle allows steam to pass from the inside of the valve through its short central channel towards the left, and thence transversing through a couple of holes into the lifter steam pipe. This will produce a delivery of water up the water pipe to the forcer, from which it will pass to the overflow through the combining tube of the forcer.

As soon as water is seen to flow from the overflow pipe, the steam handle is turned still more, until the collar on the spindle comes into contact with the slotted web of the steam valve, compelling it to move to the right, and thus uncover the steam cone of the forcer to steam. The forcer then forces the feed into the boiler.

It will be noticed that the steam valve possesses a pair of fingers which are capable of moving the forcer steam cone, thus permitting an adjustment of the water aperture into the forcer combining cone.

The overflow arrangement has to accommodate both the lifter and forcer simultaneously, and after the injector begins to feed the boiler it must close the cones to the atmosphere, because the pressure in them may exceed that of the atmosphere, on account of the high temperature of the delivery jet.

In fig. 49 the overflow is shown closed, as it is while the injector is feeding. The delivery cone is capable of sliding axially, and terminates at the overflow end in an enlargement, in which can slide the combining cone. The other end of the latter can slide freely in a corresponding enlargement of the lifting tube.

When the lifter delivers water (at starting) to the forcer, the water drives the combining and delivery cones away from the lifting tube, and at the same time separates the combining and delivery cones, thus forming a very large outlet for the water until the jet has been established in the forcer. When this occurs the pressure on the pressure end of the combining cone urges it towards the steam end, and effectively closes all the overflow openings to the atmosphere. The makers of this injector have recently altered the steam valve so that as it opens the steam cone moves inwards and adjusts the amount of feed to the forcer.

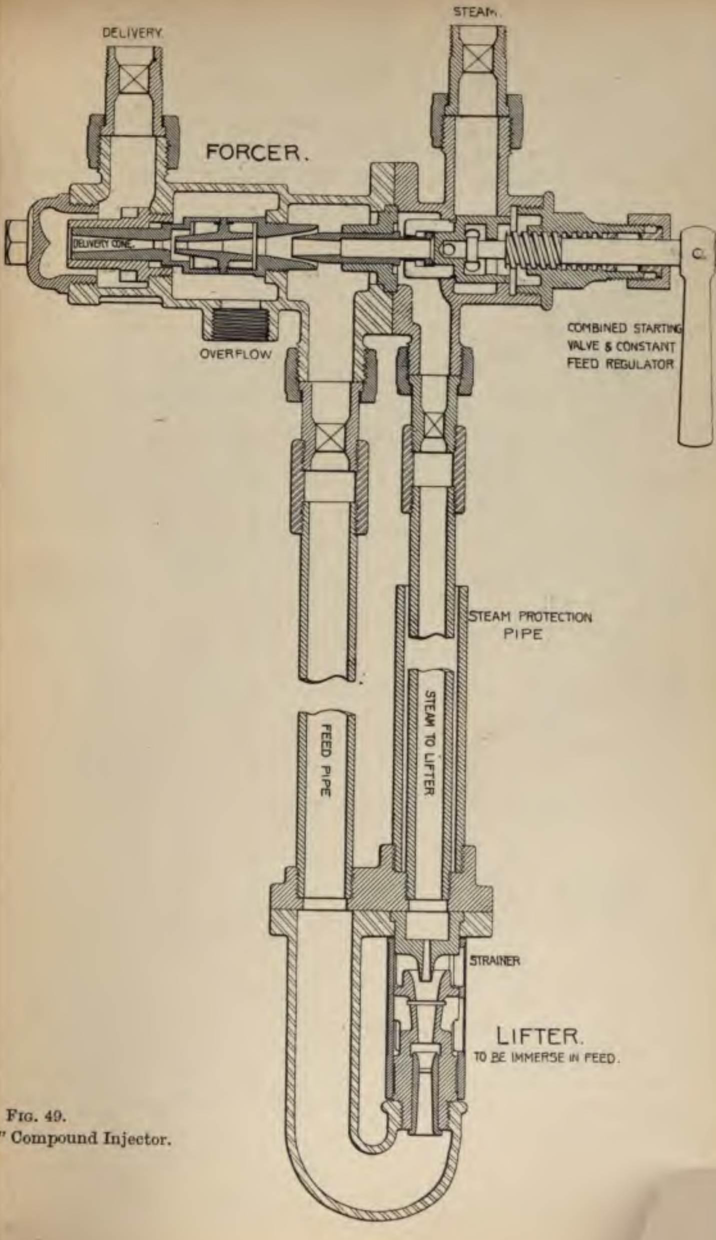


FIG. 49.
" Compound Injector.

This firm also makes a combination "Albion" injector for locomotives, two sections of which are shown in figs. 50 and 51, with a couple of elevations in fig. 53, and a

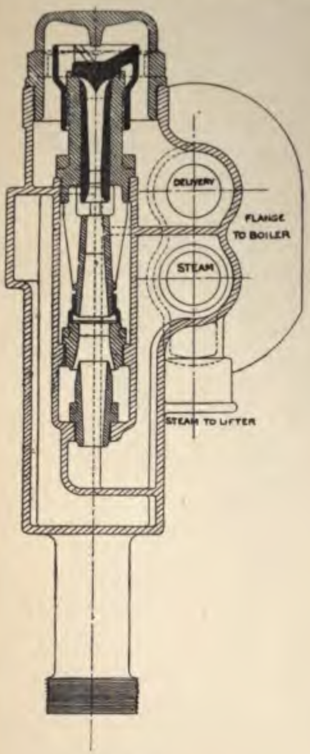


FIG. 50.

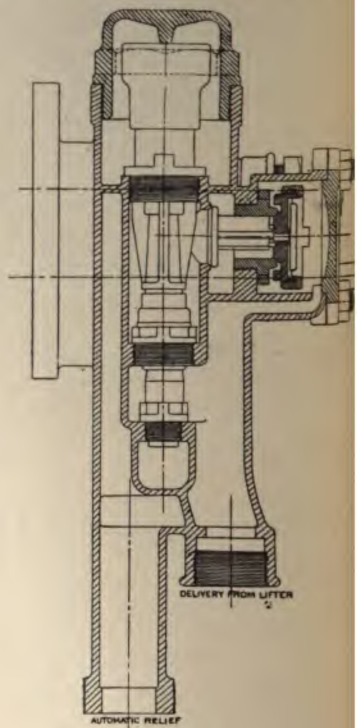


FIG. 51.

Longitudinal sections of the "Albion" Compound Injector.

general arrangement in fig. 52. The first overflow or relief orifice is covered by a lifting sleeve, which gravitates on to its seat.

These illustrations show a type of overflow valve which has now been superseded by the arrangement just described (fig. 49). It is interesting in the development

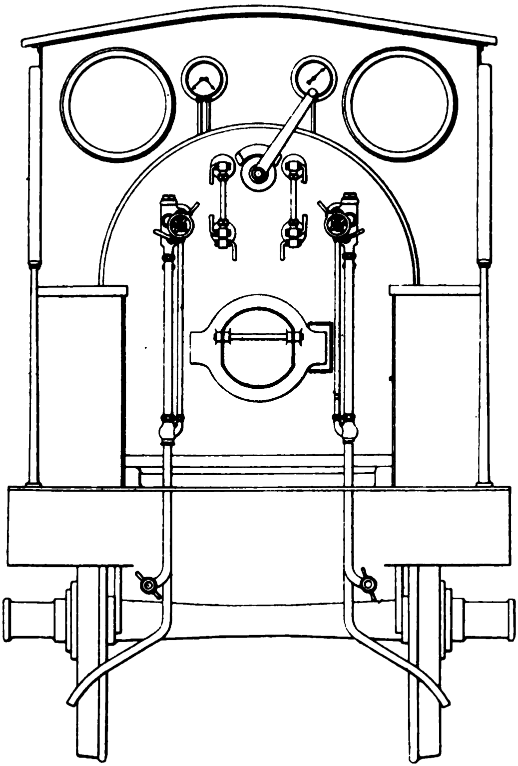


FIG. 52.—General arrangement of "Albion" Combination Injector, as fitted to Locomotives.

of this injector, and consists of two valves, one covering the orifice to the forcer overflow chamber, and the other covering the channel connecting the lifter delivery with the overflow pipe.

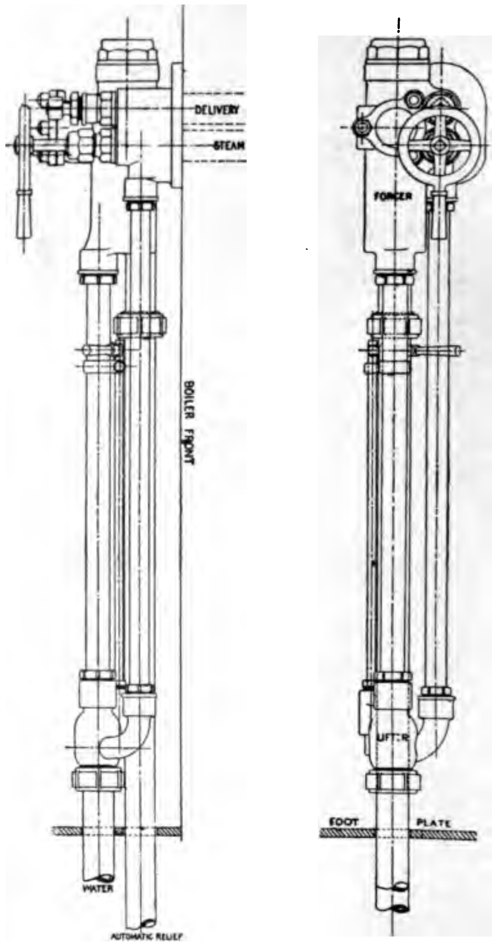


**Fig. 53.—Front and side elevation of the Combination Type of "Albion" Inje
for Locomotives.**

In the centre of the forcer overflow valve slides a pin or spindle, which passes freely through the valve shown in

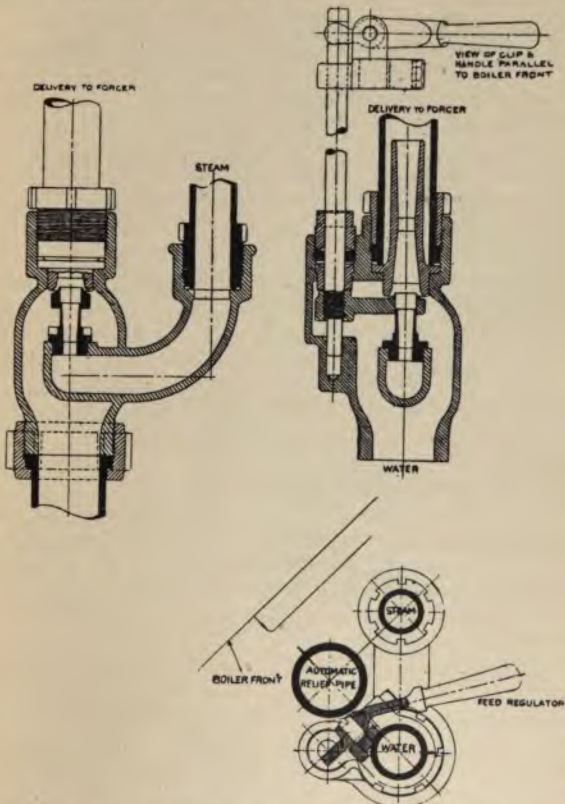


FIG. 54.—Sections of the Lifter of the "Albion" Injector, showing Water Regulator.

section, and terminates in a disc, over which is passed a flexible diaphragm, which is clamped at its edge to the rim of the valve shown in section. At starting the

pressure of the lifter delivery in the forcer overflow chamber pushed that valve off its seating, and it carried with it the valve shown in section, thus throwing both lifter and forcer in connection with the overflow.

As soon as the forcer jet was established, the pressure in the forcer overflow was relieved, and the valves were forced on to their seatings. The forcer overflow valve was held to its seat by the pressure of the lifter delivery on the disc being transmitted along its stem to the forcer valve. If wear should occur with either valve the flexible diaphragm would take it up.

Two elevations of the combination "Albion" injector, shown in figs. 50 and 51, are shown in fig. 53. The appearance is neat, and the adjustments are all made by handles at a convenient height. The water regulator in this illustration is arranged in the lifter, situated just above the footplate, two sections of which are shown in fig. 54. A small cone slides over the steam cone, and is operated by means of the handle connected to it by the vertical rod. The handle is situated just below the forcer.

In the top part of fig. 55 is a vertical section through the steam and delivery valves of this injector. The steam valve, it will be observed, admits steam to the lifter only, on being partially opened, and to both forcer and lifter when full open. The middle section in fig. 55 is a horizontal section through the overflow valve; while the bottom part of the figure is a section below the overflow valve.

A compound injector which is extremely simple in construction is shown in fig. 56. It is the invention of Mr. Strickland Kneass, of the firm of Messrs. William Sellers and Company, of Philadelphia, who first began to make injectors in America. The forcer, it will be seen, has two openings or overflows, and the lifter has one, to facilitate starting, all of them opening into the same chamber, the outlet from which is covered with a non-return valve.

The steam valve has to control the steam passing to the central forcer cone and to the annular lifter steam cone. On slightly moving the starting handle, the steam

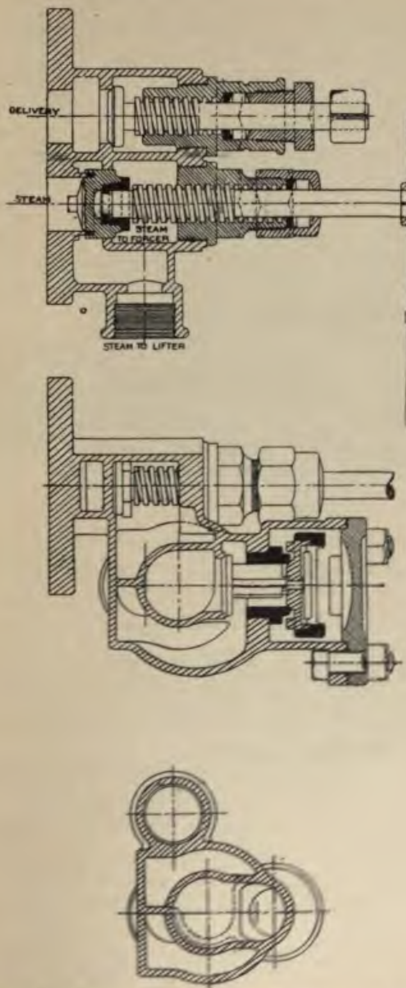


FIG. 55.—Sections of the "Albion" Combination Injector.

valve comes off its seat and allows steam to pass to the annular cone, but not to the central forcer cone.

The steam passing into the suction tube escapes easily through the overflow openings, and the feed is sucked up and escapes at the overflow. The steam handle is then moved completely over, withdrawing the plug on the end of the valve stem and permitting steam to flow down the forcer steam cone to drive the feed into the boiler.

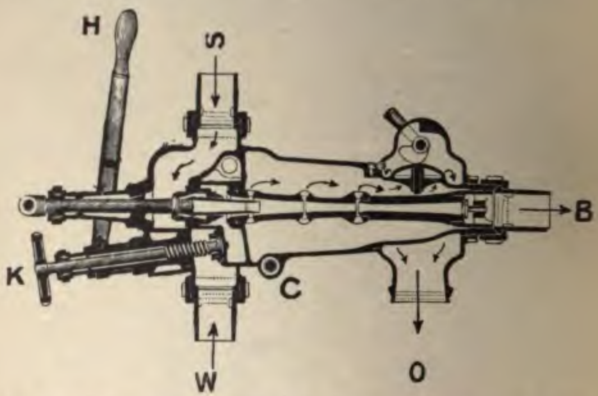


FIG. 56.—Messrs. Wm. Sellers and Co's. most recent Compound Injector.

The overflow pipe casting is slipped over the body of the instrument and held there by a nut. This injector requires no special regulation for a change of boiler pressure, and its range of capacity is over 60 per cent of maximum delivery. It can be used for heating the feed, as in locomotive practice, by throwing over the cam over the overflow valve and turning on steam, the water valve having been previously opened.

Fig. 57 shows a section through the newest type of Friedmann injector (sold in this country by Messrs. R. Klinger and Company, London). It very much resembles the one just described, but it has in addition a small chamber connected to the water pipe through the water regulator. This chamber communicates with the overflow

chamber through a non-return valve. If the forcer can take more than the lifter supplies to it, the vacuum in the overflow chamber will induce water to flow through the supplementary passage and into the cones of the forcer by means of the overflow orifices.

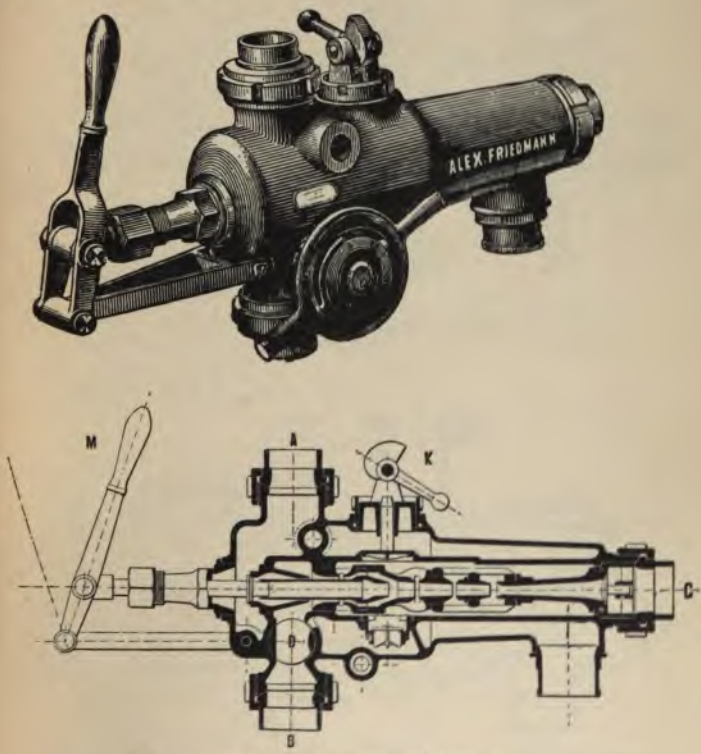


FIG. 57.—The Friedmann Compound Injector.

An adaptation of the Friedmann injector to form a horizontal combination pattern for the side of the firebox is shown in fig. 58. The steam valve V is independent of

the starting handle M. The warming valve is K, while Y is the check valve casing, S being the screw down delivery

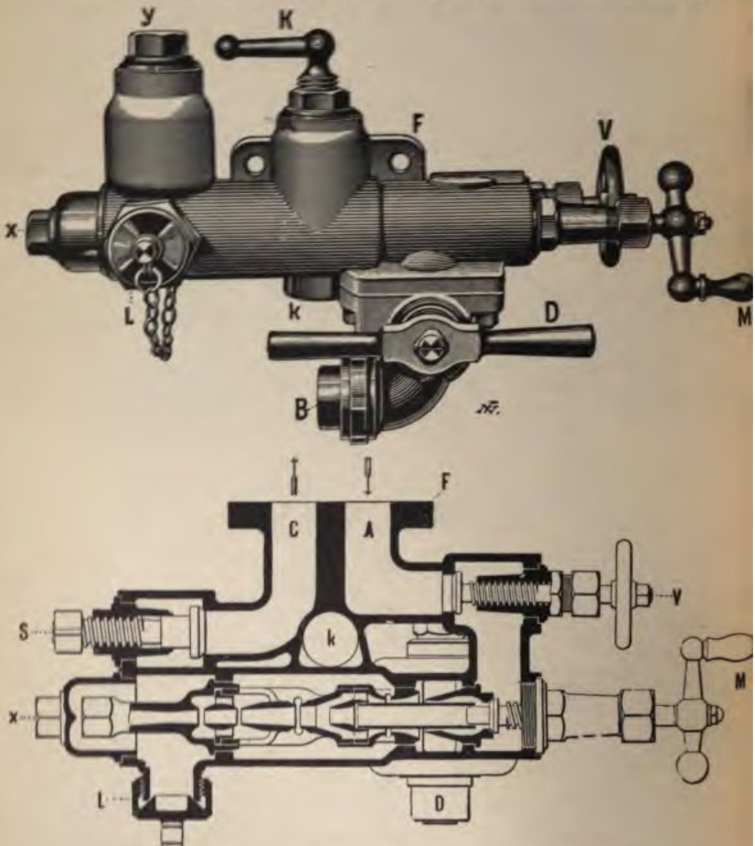


FIG. 58.—The Friedmann Injector, Horizontal Combination Type.

valve for the purpose of removing the cones under steam. The cap L is for the purpose of attaching a hose, and B is the feed connection.

The vertical combination pattern made by the same firm is shown in fig. 59. As before, S is the delivery screw

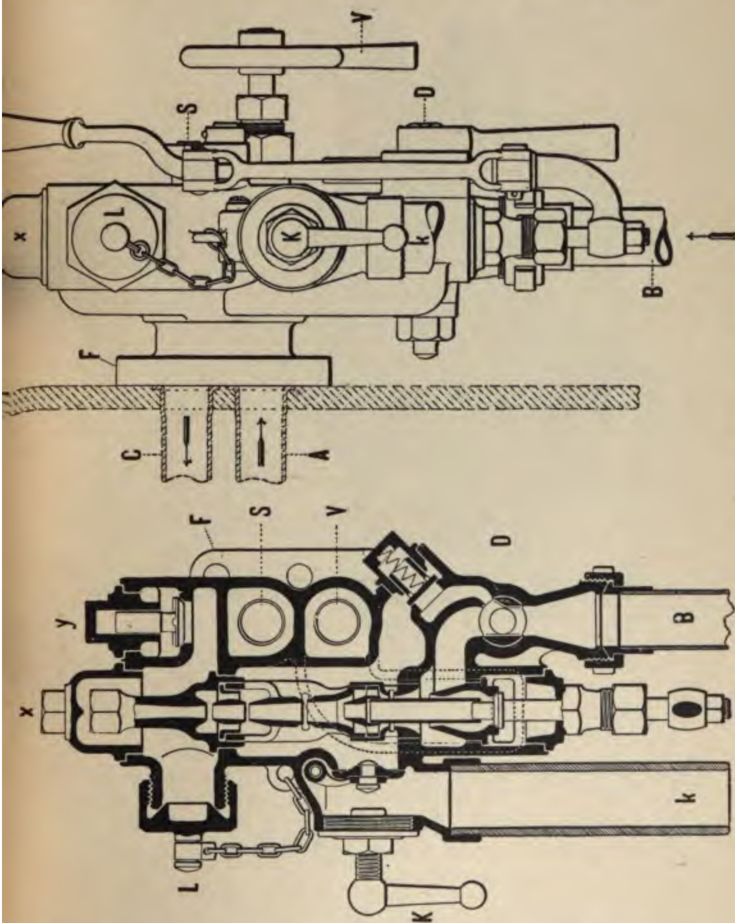


Fig. 59.—Friedmann's Vertical Combination Injector.

down valve, and V the steam valve. The starting handle M is arranged as shown *for* convenience in operation. Th

supplementary channel from the water cock is clearly shown.

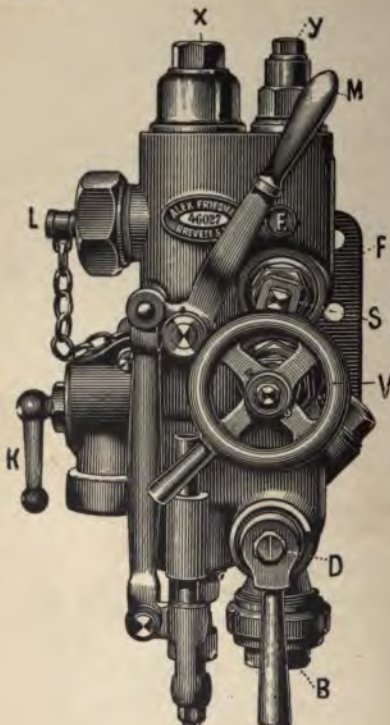


Fig. 59A.

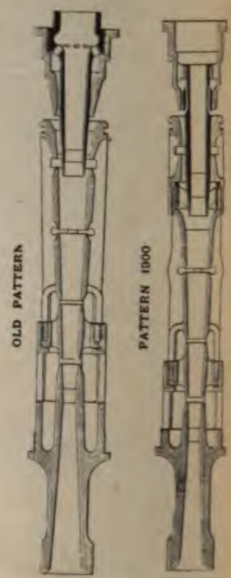


Fig. 60.

Fig. 60 shows an improvement introduced by Messrs. Friedmann in the arrangement of their cones to facilitate fixing and removal. They also put a *special metal* sleeve on the steam cone of the lifter to increase its durability.

CHAPTER IV.

Injector Calculations for Simple High-pressure Types.

To facilitate calculation, the following system of nomenclature will be adopted:—

P = pressure absolute in pounds per square foot.
p^1 = pressure absolute in pounds per square inch.
p = pressure (gauge) in pounds per square inch.
v = velocity in feet per second.
V = volume of 1 lb. of fluid in cubic feet.
ρ = density of fluid in pounds per cubic foot.
d = diameter.
W = pounds of feed taken through the suction pipe per second.
w = pounds of steam flowing through the steam pipe per second.
β = ratio $\frac{\text{water}}{\text{steam}} = \frac{W}{w}$.
A = sectional area.
H = head in feet.
d_s = the diameter of the steam cone at smallest section, in *inches*.
R = reaction of the steam jet in pounds.
λ = total heat of evaporation of 1 lb. of steam from water at freezing point.
$c = \frac{\text{delivery pressure}}{\text{boiler pressure}}$.

Suffixes—

1 refers to boiler (generally) and steam pipe, fig. 2.
2 refers to end of steam cone, fig. 2.
3 refers to end of delivery cone at the smallest section, near the overflow.
4 refers to delivery pipe.
5 refers to orifice between the steam and combining cones, through which the feed passes.
6 refers to free surface in feed-water tank.
a refers to atmosphere.
s refers to steam cone at smallest section.
d refers to delivery cone at smallest section.

The principle of momentum on which calculations are based was given in the opening chapter. It is this:—

Considering *any direction* in space, the total momentum of a system of bodies before collision equals the total momentum of the same system after collision when the momentum is measured in the above specified direction.

In the injector we have a mass of steam colliding with a mass of water, and the momentum of the steam, measured along the axis of the injector before impact, together with the momentum of the feed water, also measured in the direction of the axis, must equal the momentum of the mixed steam and water after collision, measured in the direction of the axis. The momentum of the steam jet per second as it leaves the steam cone must be

$$\frac{w\, v_2}{g} \text{ in gravitation limits.}$$

The momentum of the feed water entering the injector is necessarily very small, and can generally be neglected. Its influence will be shown later. The momentum of the combined mass of steam and water entering the delivery cone after collision is

$$(\mathrm{W} + w)\frac{v_3}{g} \text{ units.}$$

Then the equation of momentum is

$$\frac{w\, v_2}{g} + 0 = (w + \mathrm{W})\frac{v_3}{g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Now, Mr. Rosenhain has experimentally determined the left side of this equation* for steam nozzles of similar shape to those used in steam turbines, and which are very similar to those used in injectors.

Mr. Rosenhain measured the reaction caused by the efflux of steam from a nozzle at different pressures; and as the force or reaction of the jet in the chamber opposite to the jet equals the change of momentum per second of the steam in the steam jet, we can write

$$\mathrm{R} = \text{reaction or impulse in pounds} = \frac{w\, v_2}{g}.$$

* See the Proceedings of the Institution of Civil Engineers, vol. 140, page 199.

With a total taper* in the nozzle of about 1 in 12, the reaction or impulse or change of momentum per second can be represented by

$$R \text{ lbs.} = d_s^2\,(1{\cdot}08\,p_1 - 12) \quad . \quad . \quad . \quad . \quad (2)$$

It should be noted that while the diameter d_s in equation 2 refers to the smallest section or throat of the steam cone, and is measured in inches, the pressure p_1 is the gauge pressure of the boiler steam. Further, the equation does not hold much below 40 lb. per square inch.

It should also be noted that this value of the impulse or reaction is not that for frictionless flow, but is the actual value when the various resistances are taken into account, as they were in the above experiments.

We may now write equation (2) as

$$d_s^2\,(1{\cdot}08\,p_1 - 12) = R = (w + W)\,\frac{v_3}{g} \quad . \quad . \quad (2a)$$

But the mass of fluid passing through the sectional area A_s per second (the delivery)
= volume per second in cubic feet × density of fluid there
$= A_s\,v_3\,\rho_s$ lbs. $= (w + W)$ lbs. . . . . (2b)

and $A_s = \frac{\pi}{4}\,d_3^2$ square feet, if d_3 is in feet;

or $= \frac{\pi}{4} \times \frac{d_s^2}{144}$ square feet, if d_s is in inches.

Substituting in equation (2a) we get—

$$d_s^2\,(1{\cdot}08\,p_1 - 12) = \frac{A_s\,v_3^2\,\rho_s}{g}$$

$$= \frac{\pi}{4} \times \frac{d_3^2}{144} \times \frac{v_s^2\,\rho_3}{g}$$

$$= \frac{v_3^2}{2g} \times \frac{\pi\,d_3^2\,\rho_3}{288} \quad . \quad . \quad . \quad (2c)$$

---

* By total taper we mean the deviation of the cone from the cylindrical shape in a given length; thus, in fig. 61, the deviation is $d_2 - d_s$ in the length x.

Now consider the stream of fluid between the section (3) and the delivery pipe (4) in fig. 2. By Bernouilli's equation we must have

Total energy at section (3) = loss of energy between (3) and (4) + total energy at section (4);

or, when referring to 1 lb. of fluid and denoting the loss of head by h ft., we have

$$\frac{P_3^1}{\rho_3} + \frac{v_3^2}{2g} = h + \frac{P_4^1}{\rho_4} + \frac{v_4^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

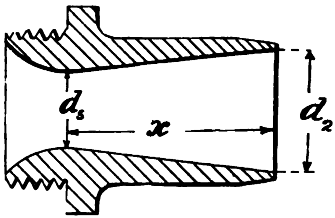


FIG. 61.

The last term in this equation is negligible for taking an actual case where $v_3 = 160$ ft. per second, and $d_3 = \cdot 3$ in., while $d_4 = 1\cdot 5$ in.; the ratio—

$$\frac{v_3}{v_4} = \frac{A_4}{A_3} = \frac{d_4^2}{d_3^2} = \frac{25}{1},$$

and
$$\frac{v_4^2}{2g} = \frac{v_3^2}{(25)^2 \times 2g} = \frac{v_3^2}{2g} \times \frac{1}{625};$$

hence $\frac{v_4^2}{2g}$ can be neglected in comparison with $\frac{v_3^2}{2g}$, and we can write zero for it. Equation (3) then becomes—

$$\frac{v_3^2}{2g} = h + \frac{P_4^1}{\rho_4} - \frac{P_3^1}{\rho_3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3a)$$

Substituting this value of v_3 in equation (2c) we get—

$$d_s^2\,(1\cdot 08\,p_1 - 12) = \left(h + \frac{P_4^1}{\rho_4} - \frac{P_3^1}{\rho_3}\right)\frac{\pi\, d_3^2\, \rho^3}{288}$$

which may be written as—

$$\left(\frac{d_s}{d_3}\right)^2 = \left(\frac{h}{144} + \frac{p_4^1}{\rho_4} - \frac{p_3^1}{\rho_3}\right)\frac{\pi\,\rho_3}{2\,(1\cdot 08\,p_1 - 12)} \quad . \quad (4)$$

Now if we knew the loss of head in passing through the delivery cone, together with the density of the stream ρ_3 at the smallest section, we could immediately calculate the ratio of the diameters and thus determine the proper size of the steam cone for any size of injector and for any particular boiler pressure. But the quantity h has so far not been determined, and very little is known of ρ_3. The value of ρ_4 must be that of hot water at the temperature of the delivery.

Before proceeding further we must examine the different conditions under which injectors are known to work, and thereby we may be able to gain some information as to the values of h and ρ_3.

There cannot be more water passing through the injector than the steam will drive into the boiler. Hence, if v_3 in equation (1) is only just large enough to produce a pressure ρ_4 in the delivery pipe, so that the water will enter the boiler,. any excess of feed water would reduce v_3, and the injector could no longer open the feed check valve, and consequently would stop working or only deliver through the overflow. Under these conditions the amount of feed water per pound of steam, or β, is a *maximum*, or we may say that *maximum β corresponds to minimum v_3*.

If β is decreased, then equation (1) shows that v_3 must increase with a constant supply of steam. But if the delivery cone is designed to take the maximum β, the decreasing of β will leave some of the section of the cone unused, if the stream lines are as perfect as was assumed to be the case with maximum β. Also, the greater velocity v_3, consequent upon the reduction of β, will permit of a further reduction in the sectional area of the delivery cone. Hence we may say that, *with a reduction in β, the delivery cone cannot be quite full of stream lines as we have assumed to be the case with maximum β.*

If the delivery cone does not run full bore, then Bernouilli's equation does not hold accurately, though equations (2), (2*a*), and (2*b*) are still valid.

What probably happens is either (1) that the stream as it leaves the combining cone is more or less in the condition of dense spray which gradually consolidates in

the delivery cone into a continuous liquid stream that completely fills the cone after some section has been reached beyond the narrowest section; or (2) even if the jet did leave the combining cone in moderately regular stream lines, the space between them and the delivery cone surface may be filled with eddies which absorb energy and may account for the loss of head h in equation (3). In the last case the channel would be practically full, and equation (3) could be used to account with very fair approximation for what really happens. In the former case the spray particles must coalesce soon after the minimum section is reached, but there must be a loss of energy due to the eddy motion in the jet. In this case also equation (3) will not be very widely out. In all probability eddy motion in the delivery cone is the chief cause of loss of head, and this may be reduced by increasing the back pressure ρ_4, but it is undesirable to increase the back pressure for the reasons previously stated.

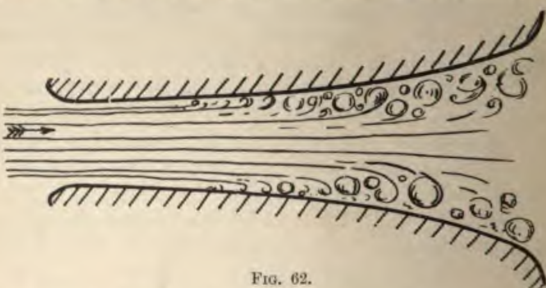


FIG. 62.

The spray mentioned above reduces the density of the stream at its entrance to the delivery cone below that of a continuous liquid, while uncondensed steam may and probably does exist at the entrance to the delivery cone. This would tend to fill the cone and gradually condense as it travelled along the cone against the increasing pressure without producing any appreciable discontinuity.

Fig. 62 gives an idea of the eddy motion suggestion, while fig. 63 represents the stream lines when working with maximum β and the cone is quite full of stream lines.

If there is any spray, the particles would not all move in an axial direction, and we should probably have the action suggested in fig. 64.

The question now arises, How much loss of head exists with different values of β, and what is the probable density ρ_3?

It would be useless to design an injector to deliver the maximum quantity of feed possible, assuming no losses,

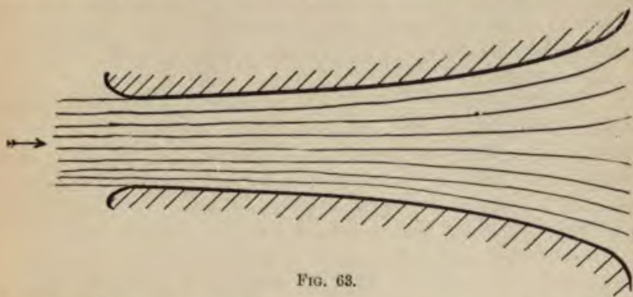


FIG. 63.

because if there occurred a slight resistance in the delivery pipe the injector would throw off.

A margin must be allowed for, to permit the injector to work after a certain deposit of sediment which occurs

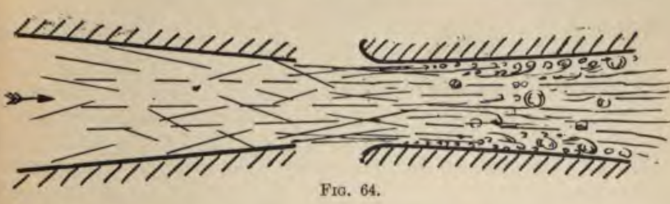


FIG. 64.

after a little time. This sediment offers a resistance to the flow of water, and consequently there must be a certain etxra pressure produced at the end of the delivery cone to overcome the resistance; in other words, the injector must be designed to feed (when necessary) against a higher pressure than that in the boiler. That this excess of

pressure is possible is easily demonstrated by inserting a valve in the delivery and throttling the discharge; that is, increasing the resistance to the flow of the water. By this means an injector taking steam at, say, 120 lb. per square inch may be made to deliver water against a pressure of 200 lb. per square inch. This can be done without the flow of steam being altered in any way, and therefore without v_3 and β being altered.* Hence the difference must be in the loss of head which takes place when feeding against any pressure lower than the maximum—the lower the pressure, the greater the loss of head.

It would then seem rational to settle upon the excess of pressure above that of the boiler, which the injector should be capable of delivering against, and then, in the absence of further definite information, to assume that the loss of head at this pressure is zero or some small quantity. If $h = o$ the stream is practically perfect, and if the stream is wholly condensed and there is no spray the density ρ_3 must be that of hot water. The excess of pressure which seems reasonable to permit is from 50 to 75 per cent of the boiler pressure; and this will be found to coincide with general practice.

Returning now to equation (4), we can calculate the values of the ratio

$$\frac{d_s}{d_3}$$

for different values of the excess of delivery over boiler pressure; *on the assumption that at these excess pressures there is no loss of head* h *and the density of the delivery stream* ρ_3 *is that of hot water, or somewhere about 60 lb. per cubic foot.*

The above table has been calculated on the assumption that the overflow orifice was open to the atmosphere, and therefore the pressure there was about 15 lb. per square inch.

---

* There may be a small change in β, because the increase in pressure $p_4{}^1$ may produce an increase in pressure $p_3{}^1$, and this may affect the difference of pressure driving the water into the combining cone from the suction pipe.

Gauge-pressure of boiler, lbs.	[illegible]				
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
100	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
200	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
300	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

If the density [illegible] table should be multiplied by [illegible]

ρ_2 being in pounds per cubic foot.

[illegible]

The weight of water and condensed [illegible] passing through the delivery tube per second [illegible]

This must equal A_3 [illegible]

hence $$v_3 = \frac{w + W}{A_3 \rho_3}$$

Making this substitution in equation [illegible], page [illegible], together with

$$\frac{\pi}{4}\frac{d_3^2}{144}$$

for A_3, d_3 being measured in inches, we get

$$d_t^2(1\cdot08\,p_1 - 12) = \frac{(w + W)\,576}{g\,\rho_3 \pi\, d_3^2}$$

and

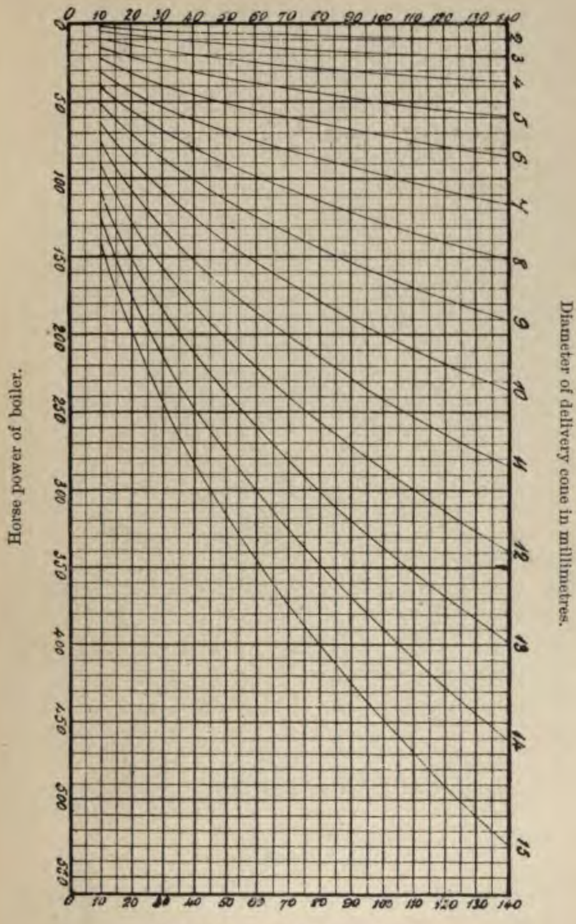
$$(w + W) \text{ lbs. per sec.} = d_t\, d_3 \sqrt{\frac{(1\cdot08\,p_1 - 12)\,g\,\rho_3 \pi}{576}}$$

Maximum weight of feed water delivered per hour in pounds when the overflow is open. The steam for the injector is taken from the same boiler into which the feed water is delivered.

Size of the combining cone orifice in millimetres.	Size of the combining cone orifice in inches.	Pressure of steam in pounds per square inch.												
		20	30	40	50	60	70	80	90	100	110	120	130	140
		Delivery of feed water in pounds per hour.												
2	·0787	350	430	500	540	610	660	710	750	800	830	870	910	930
3	·1181	800	970	1120	1260	1380	1500	1600	1700	1780	1870	1960	2030	2110
4	·1574	1420	1730	2010	2250	2460	2660	2850	3010	3170	3330	3480	3620	3760
5	·1968	2220	2720	3130	3510	3850	4160	4450	4710	4960	5220	5450	5600	5870
6	·2362	3200	3920	4520	5060	5550	5980	6400	6780	7150	7500	7830	8160	8460
7	·2755	4350	5330	6160	6880	7550	8150	8710	9210	9730	10210	10670	11100	11520
8	·3148	5680	6960	8030	8980	9850	10620	11370	12060	12720	13350	13930	14500	15050
9	·3543	7200	8820	10170	11380	12470	13470	14400	15270	16100	16880	17630	18350	19050
10	·3936	8880	10880	12560	14050	15400	16630	17770	18870	19870	20860	21770	22660	23520
11	·4330	10750	13170	15200	17000	18630	20120	21500	22810	24050	25220	26330	27420	28460
12	·4724	12800	15670	18100	20230	22170	23950	25600	27150	28610	30020	[illegible]	32330	33870
13	·5127	15010	18400	21230	23520	26020	28100	30050	31870	33580	35230	[illegible]	38300	39750
14	·5510	17410	21330	24620	27530	30180	32610	34850	36960	38950	40870	42670	44410	46100
15	·5904	19980	24500	28270	31610	34650	37420	40000	42420	44710	46910	49000	50980	52920
16	·6296	22750	27870	32170	35970	39420	42570	45510	48270	50870	53370	55750	58010	60220
17	·6691	25670	31460	36310	40610	44500	48060	51380	54500	57430	60260	62910	65480	67980
18	·7086	28780	35270	40710	45520	49900	53880	57600	61100	64380	67560	70550	73420	76330
19	·7479	32070	39300	45360	50720	55600	60030	64180	68070	71750	75270	78610	81800	84920
20	·7872	35530	43550	50260	56210	61600	66520	71100	75420	79500	83410	87100	90610	94100
21	·8266	39180	48030	55410	61960	67910	73330	78100	83860	87600	91670	95410	98470	101860
22	·8660	43000	52680	60810	68010	74520	80480	85710	92050	96150	100610	104710	108070	111800
23	·9054	46830	57420	66380	74120	81460	87570	[illegible]	[illegible]	105080	109970	114860	118120	122180
24	·9448	51000	62520	72280	80710	88700	95350	102000	108210	114420	119750	125060	128600	133050
25	·9841	55330	67850	78430	87580	96250	103460	110680	117420	124150	129980	135710	139560	144370

Note. —The nominal horse power multiplied by 100 gives approximately the number of pounds of feed water per hour.

Pounds pressure per square inch.



Pounds pressure per square inch.

Fig. 65.

These are represented approximately by the expression—

$$\text{lbs. water per hour} = 20\ d_2^{\,2}\ \sqrt{p_1} \quad . \quad . \quad . \quad (6b)$$

where d_3 is measured in millimetres and p_1 is the boiler gauge pressure in pounds per square inch.

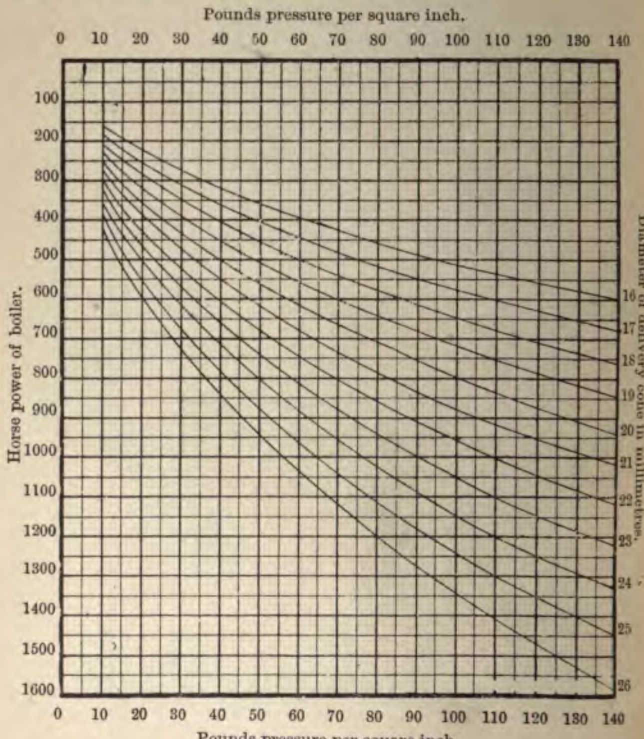


FIG. 66.

Recapitulating, the relation indicated by equation (1) must hold on all occasions where steam collides with water. Also with a given p_1, the impulse

$$\frac{w\ v}{g}$$

must be constant or nearly so in an injector, and therefore $(1 + \beta)\, v_3$ must be constant. That is, if β is increased, v_3 must decrease correspondingly.

This may go on until the velocity of the jet is insufficient to enable it to enter the boiler. This is governed by equation (3). If the injector is not feeding a boiler, but delivering at some *lower* pressure, the value of β may be still further increased, and as a result v_3 will decrease.

We see then that there is a maximum value for β when the injector is feeding a boiler, but a higher value of β is possible if the injector is delivering at a lower pressure than that of the boiler.

In the preceding pages the diameter of the steam cone has been obtained in equation (4) and given in the table on page 75 by considering the impact equation (1), the equation of continuity (2*b*), Bernouilli's equation (3), and an experimental value of the impulse of the steam jet.

Bernouilli's equation imposes the condition of a certain delivery pressure. This is not altogether necessary if β happens to be known or is decided upon.

Thus if we substitute the value of v_3 from equation (2*b*), in equation (2*a*) we get

$$d_s^2\,(1 \cdot 08\, p_1 - 12) = \frac{(w + \mathrm{W})^2}{g} \times \frac{576}{\pi\, \rho_3\, d_3^2}.$$

If we write

$$w + \mathrm{W} = w\left(1 + \frac{\mathrm{W}}{w}\right) = w\,(1 + \beta)$$

and substitute the value of w, deducted from the results of Mr. Rosenhain's experiments, namely,

$$w = d_s^2\,(\cdot 0113\, p_1 + \cdot 175) \text{ lb. per sec.}$$

we get

$$\frac{d_s}{d_3} = \frac{1}{(1+\beta)\,(\cdot 0113\, p_1 + \cdot 175)} \sqrt{\frac{g\, \pi\, \rho_3\,(1 \cdot 08\, p_1 - 12)}{576}}. \quad (7)$$

This equation is independent of the back pressure ρ_4. From it we gather that the steam cone diameter must be

increased with an increase in the density ρ_3 of the delivery jet, and decreased with an increase in β, which means that less steam is required because only a *part* of the previous

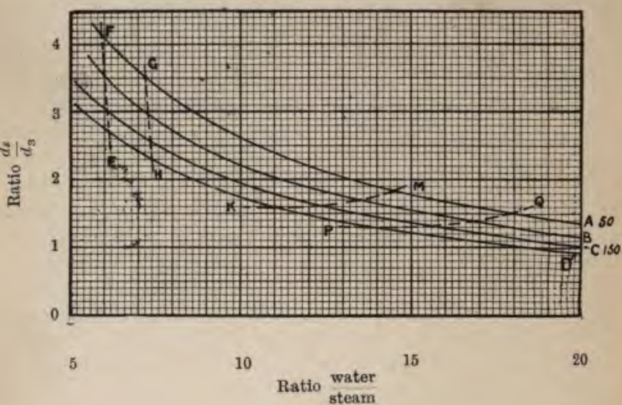


FIG. 67.

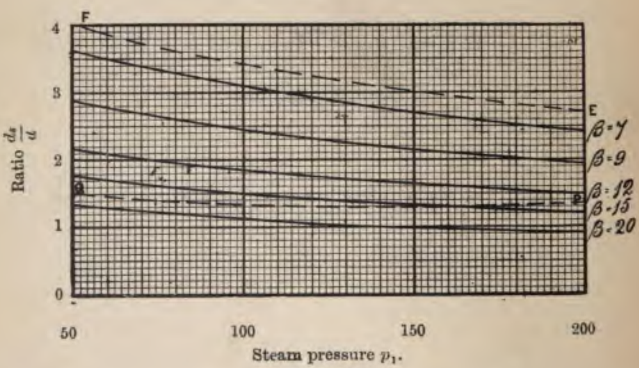


FIG. 68.

delivery can get through the delivery cone with the reduced velocity corresponding to the increase of β. To put it in another way, if the steam cone is maintained

constant, then by equation [illegible] d [illegible] β to accommodate the increase [illegible] smaller velocity.

As in the previous calculations [illegible] cubic foot. If the true density [illegible] this, then multiply the number [illegible]

and the true ratio will [illegible]

Substituting 50 [illegible] respectively for p [illegible]

$$\frac{d_1}{d_2} = \frac{[illegible]}{[illegible]} \text{ for 50 lb. pressure [illegible]}$$

$$= \frac{[illegible]}{[illegible]} \text{ for 100 lb.}$$

$$= \frac{[illegible]}{[illegible]} \text{ for 150 lb.}$$

$$= \frac{[illegible]}{[illegible]} \text{ for 200 lb.}$$

By giving β a few values such as those [illegible] sufficient data is available to plot the curves [illegible] and 67.

β	$p_2 = 50$	$p_2 = 100$	$p_2 = 150$
7	3·5[illegible]	[illegible]	2·[illegible]
9	2·85	[illegible]	2·[illegible]
12	2·1[illegible]	[illegible]	[illegible]
15	1·2[illegible]	[illegible]	[illegible]
20	1·[illegible]	[illegible]	[illegible]

The numbers in this table can be represented almost exactly by the expression

$$\frac{d_s}{d_3} = \frac{40}{(1+\beta)}\sqrt{\frac{\rho_3}{p_1+60}} \quad . \quad . \quad . \quad (8)$$

when $\rho_3 = 60$ lb. per cubic foot.

Working Limits.

As β lb. of water at t_5 deg. Fah. are mixed with 1 lb. of steam whose total heat $(S_1 + x_1 L_1)$ may be called λ, and resulting in a final temperature t_3 of the mixture, then the heat given up by the steam must be received by the β lb. of water, or—if S represents $(t-32)$

$$w(\lambda - S_3) = W(t_3 - t_5)$$

or,

$$\beta = \frac{\lambda - S_3}{t_3 - t_5}$$

and

$$1 + \beta = \frac{\lambda - S_3}{S_3 - S_5} + 1 = \frac{\lambda - S_5}{S_3 - S_5}$$

$$= \frac{\lambda - S_5}{t_3 - t_5}.$$

Now if the values of λ be taken from the steam tables and plotted on a gauge pressure base, no pressures below 50 lb. per square inch being considered, it will be found that

$$\lambda = 1164 + {\cdot}2p_1, \text{ very nearly} \quad . \quad . \quad . \quad (9)$$

then

$$1 + \beta = \frac{1164 + {\cdot}2p_1 - (t_5 - 32)}{t_3 - t_5}$$

$$= \frac{1196 + {\cdot}2\,p_1 - t_5}{t_3 - t_5} \quad . \quad . \quad . \quad (10)$$

Now if β is large the rise of temperature $t_3 - t_5$ will be correspondingly small; also minimum β corresponds to a maximum rise of temperature.

Taking the lowest possible [illegible] 32 deg. Fah., and assuming that the [illegible] 212 deg. Fah.,* we have from [illegible]

$$\text{minimum} \left([illegible] \right) = [illegible]$$

$$= [illegible]$$

By inserting different [illegible]

p_1	[illegible]	[illegible]
50	[illegible]	[illegible]
100	[illegible]	[illegible]
150	[illegible]	[illegible]
200	[illegible]	[illegible]

An average of [illegible] represented [illegible]

$$\frac{[illegible]}{\text{rise of temperature of [illegible]}} \qquad [illegible]$$

As previously [illegible] hold on all occasions. [illegible] to permit the jet [illegible] with it the maximum [illegible] Hence, for maximum [illegible] must be a minimum and [illegible] must be a minimum or

$$s - \frac{y}{[illegible]} - [illegible]$$

must be a minimum. [illegible]

$$[illegible]$$

* This is not an absolute maximum [illegible] injector.

as explained on page 76, the last term being negle
Substituting in equation (1), we have

$$v_2 = (1 + \beta)\ v_3 = (1 + \beta)\sqrt{\frac{P^1_1}{\rho_4} \times 2g}$$

and substituting for v_2 the value derived from Mr. R
hain's experiments, namely,

$$v_2 = g\left(\frac{1{\cdot}08\,p_1 - 12}{{\cdot}0113\,p_1 + {\cdot}175}\right)$$

we have

$$1 + \beta = \frac{v_2}{\sqrt{\frac{c\,2\,g\,P^1_1}{\rho_4}}} = \frac{g\left[\frac{1{\cdot}08\,p_1 - 12}{{\cdot}0113\,p_1 + {\cdot}175}\right]}{\sqrt{\frac{c\,2\,g\,P^1_1}{\rho_4}}} \quad . \ . \ (1$$

Then β is a maximum when $c\,P^1_1$ is a minimum.
will be the case in an injector when $c = 1$, but in
case of the steam jet pump, when $c\,p_1$ is the actual
pressure against which it is forcing water, whic
generally only a few pounds per square inch.

The right-hand side of the last equation can be si
fied by writing ·0113 $(p_1 + 15)$ for ·0113 p_1 + ·175
144 $(p_1 + 15)$ for P^1_1. Then—

$$1 + \beta = \sqrt{\frac{g\,\rho_4}{228c\,(p_1 + 15)}}\left[\frac{1{\cdot}08\,p_1 - 12}{{\cdot}0113\,(p_1 + 15)}\right]$$

and assuming $\rho_4 = 60$ lb. per cubic foot, which corresp
to perfect condensation, and therefore to maximum capa
we have for the injector

$$1 + \beta = \sqrt{\frac{230}{c\,(p_1 + 15)}}\left[\frac{1{\cdot}08\,p_1 - 12}{(p_1 + 15)}\right]$$

$$= \frac{230\,(1{\cdot}08\,p_1 - 12)}{(p_1 + 15)^{\frac{3}{2}}\sqrt{c}} \quad . \ . \ . \ . \ . \ . \ (1$$

and for the steam jet pump delivery against a pres
p_4 lb. per square inch by the gauge

$$1 + \beta = \frac{230\,(1{\cdot}08\,p_1 = 12)}{(p_1 + 15)\sqrt{p_4 + 15}} \quad . \ . \ . \ . \ . \ (12$$

[illegible]

If it [illegible]

maximum [illegible]

[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]

For the [illegible]

[illegible]

The [illegible]

increased with an increase in the density ρ_3 of the delivery jet, and decreased with an increase in β, which means that less steam is required because only a *part* of the previous

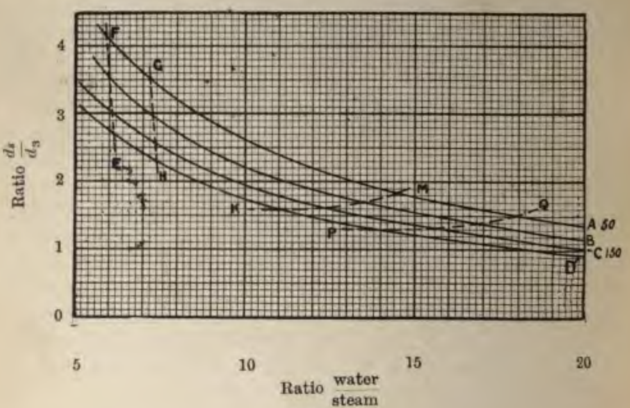


FIG. 67.

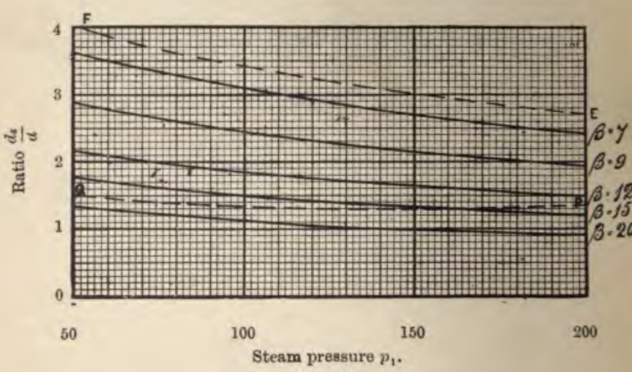


FIG. 68.

delivery can get through the delivery cone with the reduced velocity corresponding to the increase of β. To put it in another way, if the steam cone is maintained

cold water. Now, because the momentum before impact = momentum after impact, we have—

> Momentum of steam + equivalent momentum of feed water = momentum of combined steam and feed water on issuing from orifice A,

or $\frac{1}{g} \times v_2 + (1 + \beta) \frac{H}{v_3} = $ [illegible]

which simplifies to—

$$v_2 = (1 + \beta) \left[\text{[illegible]} \right]$$

Transposing, we get

$$1 + \beta = \text{[illegible]} \qquad \text{[illegible]}$$

and substitute for v_2 from the equation [illegible], and for v_3 from the equation [illegible]. We find that

$$1 + \beta = \frac{g\left[\text{[illegible]}\right]}{\text{[illegible]}} \qquad \text{[illegible]}$$

This is the same as (11) except for the extra term in the denominator to represent the suction head.

Assuming that $c = 1.5$ and $p_4 = 50$ then

when $p_1 = 50$ lb. per sq. in. and H = 0, $1 + \beta = 14.5$
and when $p_1 = 50$ lb. per sq. in. and H = 20 ft., $1 + \beta = 14.0$
also when $p_1 = 200$ lb. per sq. in. and H = 0, $1 + \beta = 11.0$
and when $p_4 = 200$ lb. per sq. in and H = 20 ft., $1 + \beta = 10.9$

Hence the dynamical influence of the suction head is negligible. The chief cause of the decrease in delivery with a large suction head is that the atmosphere has as much as it can do to lift the feed up the suction pipe, and there is therefore insufficient residual energy to give it the same velocity as with no suction head.

The reduction in delivery depends upon the temperature of the suction water. Messrs. Holden and Brooke state that with 100 lb. steam pressure the following reduction in delivery may be expected:—

Lift in feet.		5	9	13	15
Percentage reduction in delivery.	$t_5 = 50°$ Fah.	3	5·5	12·5	18
	$t_5 = 80°$ Fah.	8	11·5	19	25

The effect of temperature on the capacity of the injector is approximately as follows—for a boiler pressure of 100 lb. per square inch.

Temperature of feed $t_5°$ Fah.	50°	60°	70°	80°	90°	100°	110°	120°
Percentage reduction of delivery.	0	1·5	3	5	7·5	10·5	13·5	19·5

The following are the highest temperatures of feed with which most single tube injectors can be relied upon to work satisfactorily:—

Boiler pressure.	35	100	250
Maximum feed temperature, Fah.	150°	120°	80°

CHAPTER V.

The Steam Jet.

The velocity of steam issuing from a nozzle has been investigated experimentally by Mr. Rosenhain, Prof. Rateau, Prof. Stodola, and others.

In the first-named experiments, a steel tube was rigidly fixed at one end, and to the other was attached a small

chamber containing an opening at right angles to the length of the tube. When steam was turned on to the tube, it flowed through the said opening, and the reaction of the jet upon the side of the chamber opposite to the opening was measured by balancing it with weights, so that the tube was thereby brought back to its zero position. A pressure gauge on the chamber indicated the pressure there.

In this way the value of the *reaction* was found for a variety of pressures and nozzles, with the following average result:—

$$R = d_2 \ (1{\cdot}08p - 11{\cdot}5) \quad . \quad . \quad . \quad . \quad . \quad (15)$$

where R = reaction in pounds;
d = diameter of nozzle at minimum section in inches;
and p = gauge pressure in pounds per square inch.

The above expression does not hold good much below 40 lb. per square inch. The nozzles experimented upon had a taper of from 1 in 12 to 1 in 30. The former taper is that often used in injectors. The ratio length to diameter in this case was about 3·5.

During these experiments the weight of steam discharged was also measured, and the following equation represents the results with fair consistency above 40 lb. per square inch.

$$w \text{ lb. per sec.} = d^2 \ ({\cdot}0113p + {\cdot}175 \quad . \quad . \quad (16)$$

the symbols indicating the same quantities as before.

If the absolute pressure is used instead of the gauge pressure and we represent the former by p^1, then $p^1 = p + 15$ approximately, and the above becomes

$$w = d^2 \times {\cdot}0113 \ p^1 \text{ nearly.} \quad . \quad . \quad (16a)$$

This is almost identical with Napier's expression—

$$w = \frac{p^1 \ A}{70} \quad . \quad . \quad . \quad . \quad . \quad . \quad (16b)$$

where A = area of smallest section.

Entropy of Water, from 32 deg. Fah. to 400 deg. Fah., calculated from the formulæ, $\phi_{\tau_1} - \phi_{\tau_0} = s(\log_e \tau_1 - \log_e \tau_0)$.

Temperature. Deg. Fah. t.	Absolute temperature. τ.	Specific heat. s.	Increase of entropy per 10 deg. Fah. $\phi_{\tau_1} - \phi_{\tau_0}$.	Total entropy above water at 32 deg. Fah. ϕ_w.	Difference of entropy per 1 deg. Fah. $\Delta\,\phi_w$.
32	492	..	....	....	0·00202
40	500	1·0	0·01613	0·01613	0·00198
50	510	1·0	0·01980	0·03593	0·00194
60	520	1·0	0·01942	0·05535	0·00191
70	530	1·001	0·01906	0·07441	0·00187
80	540	1·001	0·01871	0·09312	0·00184
90	550	1·002	0·01838	0·11150	0·00181
100	560	1·002	0·01806	0·12956	0·00177
110	570	1·003	0·01775	0·14731	0·00174
120	580	1·004	0·01745	0·16476	0·00172
130	590	1·004	0·01716	0·18192	0·00169
140	600	1·005	0·01689	0·19881	0·00166
150	610	1·006	0·01662	0·21543	0·00164
160	620	1·007	0·01637	0·23180	0·00161
170	630	1·008	0·01613	0·24793	0·00159
180	640	1·009	0·01589	0·26382	0·00157
190	650	1·010	0·01566	0·27948	0·00154
200	660	1·011	0·01544	0·29492	0·00152
210	670	1·012	0·01522	0·31014	0·00150
220	680	1·013	0·01501	0·32515	0·00148
230	690	1·014	0·01480	0·33995	0·00146
240	700	1·015	0·01461	0·35456	0·00144
250	710	1·017	0·01443	0·36899	0·00142
260	720	1·019	0·01425	0·38324	0·00141
270	730	1·021	0·01408	0·39732	0·00139
280	740	1·022	0·01391	0·41123	0·00137
290	750	1·024	0·01374	0·42497	0·00136
300	760	1·026	0·01358	0·43855	0·00134
310	770	1·027	0·01343	0·45198	0·00133
320	780	1·029	0·01328	0·46526	0·00131
330	790	1·031	0·01313	0·47839	0·00130
340	800	1·032	0·01298	0·49137	0·00128
350	810	1·034	0·01284	0·50421	0·00127
360	820	1·036	0·01271	0·51692	0·00126
370	830	1·038	0·01258	0·52950	0·00124
380	840	1·040	0·01245	0·54195	0·00123
390	850	1·042	0·01233	0·55428	0·00122
400	860	1·044	0·01221	0·56649	

Entropy of Dry Saturated Steam, from 32 deg. Fah. to 400 deg. Fah.

Temperature Deg. Fah.	Absolute temperature.	Latent heat.	Entropy of 1 lb. steam.	Difference of entropy per 1 deg. Fah.	Total entropy of water + steam. Per lb.	Difference per 1 deg. Fah.
$t.$	$\tau.$	L.	$\phi s = \frac{L}{\tau}.$	$\Delta\ \phi s.$	$\phi w. + \phi s.$	$\Delta\ \phi w + s.$
32	492	1091·7	2·2189	0·00581	2·2189	0·00380
40	500	1086·2	2·1724	0·00561	2·1885	0·00363
50	510	1079·3	2·1163	0 00542	2·1522	0·00348
60	520	1072·3	2·0671	0·00521	2·1174	0 00330
70	530	1065·3	2·0100	0·00502	2·0844	0·00315
80	540	1058·3	1·9598	0·00483	2·0529	0·00299
90	550	1051·3	1·9115	0·00465	2·0230	0·00285
100	560	1044·36	1·8649	0·00449	1·9945	0·00272
110	570	1037·39	1·8200	0·00434	1·9673	0·00259
120	580	1030·42	1·7766	0·00420	1·9414	0·00249
130	590	1023·40	1·7346	0·00406	1·9165	0·00237
140	600	1016·39	1·6946	0·00393	1·8928	0·00227
150	610	1009·38	1·6547	0·00379	1·8701	0·00216
160	620	1002·37	1·6167	0·00368	1·8485	0·00207
170	630	995·33	1·5799	0·00357	1·8278	0·00198
180	640	988·30	1·5442	0·00346	1·8080	0·00189
190	650	981·24	1·5096	0·00336	1·7891	0·00182
200	660	974·18	1·4760	0·00326	1·7709	0·00174
210	670	967·10	1·4434	0 00316	1·7535	0·00166
220	680	960·03	1·4118	0·00307	1·7369	0·00158
230	690	952·94	1·3811	0·00299	1·7211	0·00153
240	700	945·83	1·3512	0·00292	1·7058	0·00148
250	710	938·76	1·3220	0·00282	1·6910	0·00140
260	720	931·56	1·2938	0·00275	1·6770	0·00134
270	730	924·41	1·2663	0·00268	1·6636	0·00129
289	740	917·25	1·2395	0·00261	1·6507	0·00123
290	750	910·06	1·2134	0·00254	1·6384	0·00119
300	760	902·86	1·1880	0·00248	1·6265	0·00113
310	770	895·64	1·1632	0·00242	1·6152	0·00109
320	780	888·41	1·1390	0·00236	1·6043	0·00105
330	790	881·15	1·1154	0·00231	1·5938	0·00101
340	800	873·38	1·0923	0·00225	1·5837	0·00097
350	810	866·58	1·0698	0·00219	1·5740	0 00092
360	820	859·27	1·0479	0·00215	1·5648	0·00089
370	830	851·95	1·0264	0·00210	1·5559	0·00085
380	840	844·58	1·0054	0·00205	1·5474	0·00082
390	850	837·20	0·9849	0·00200	1·5392	0·00078
400	860	829·84	0·9649		1·5314	

Hence the shaded area

$$= \left[\frac{T_B - T_D}{T_B + T_D} + \frac{x\,L_B}{T_B}\right](T_B - T_D) \quad . \quad . \quad (18)$$

= heat equivalent of the kinetic energy of 1 lb. of steam as it issues from the nozzle without encountering any resistance in the nozzle.

$$= \frac{1}{778} \times \frac{v^2}{2\,g}$$

At 100 lb. gauge pressure, assuming x to be unity, and the jet issuing into the atmosphere, we have

$$v = 2{,}730 \text{ ft. per sec.}$$

From Rosenhain's results we have, by using equation (17) or (17a), $v = 2{,}430$ ft. secs.

The difference, namely, 300 ft. per sec., is due to friction and eddy resistance in that particular nozzle.

It is sufficiently accurate in equation (18) to neglect the first term, and then we get

$$v = 224\sqrt{x\,L_B\,\frac{T_B - T_D}{T_B}} \quad . \quad . \quad . \quad (19)$$

At 100 lb. gauge pressure this expression gives

$$v = 2{,}630 \text{ ft. per sec.}$$

The above is sometimes put in a slightly different way, thus—

It is assumed that the expansion is adiabatic, and that all the available energy between the given temperatures is converted into kinetic energy. Hence the kinetic energy of 1 lb. must be represented by the shaded area of the indicator diagram, fig. 69, which is

Heat supplied up to B − heat rejected at D.

$$= S_B + x_B\,L_B - (S_D + x_D\,L_D) \quad . \quad . \quad (20)$$

If the pressures or temperatures corresponding to B and D are known, the only undetermined quantity in this

Differentiating, we have—

$$d\,P = n\,K\,\rho^{n-1}\,d\,\rho,$$

and

$$\frac{d\,P}{\rho} = \frac{n\,K^{n-1}\,d\,\rho}{\rho} = n\,K\,\rho^{n-2}\,d\,\rho.$$

After integrating, we obtain—

$$\int \frac{d\,P}{\rho} = \int n\,K\,\rho^{n-2}\,d\,\rho = \frac{n}{n-1}\,K\rho^{n-1}.$$

Multiply numerator and denominator of right-hand side by ρ; then, remembering that $P = K\,\rho^n$, we obtain

$$\int \frac{d\,P}{\rho} = \frac{n}{n-1}\,K\frac{\rho^n}{\rho} = \frac{n}{n-1}\,\frac{P}{\rho}.$$

Putting in the limits, and substituting the results in equation (24), we find that

$$\frac{n}{n-1}\left[\frac{P_1}{\rho_1} - \frac{P_2}{\rho_2}\right] = \frac{v_2^2 - v_1^2}{2\,g} \quad . \quad . \quad (25)$$

Let the initial velocity v_1 be zero and put in the volumes for the inverse of the densities, then (25) becomes

$$\frac{n}{n-1}\left[P_1\,V_1 - P_2\,V_2\right] = \frac{v_2^2}{2\,g} \quad . \quad . \quad (25a)$$

Equations (25) and (25a) can be expressed as

$$\frac{v^2}{2\,g} = \frac{n}{n-1}\,144\ p^1{}_1\ V_1\left[1 - \left(\frac{p^1}{p^1{}_1}\right)^{\frac{n-1}{n}}\right] \quad . \quad . \quad (25b)$$

That is, the velocity increases as p^1 decreases. In the previous calculations p^1 the exhaust pressure has been assumed to be that of the atmosphere. If the pressure at the end of the nozzle is known, it can be inserted in equation (25b) and the corresponding velocity be found on the assumption of no loss due to friction, eddies, etc. It has been found by means of very fine exploring tubes such as that at T in fig. 70, the other end of which is connected to a pressure

gauge after passing through a stuffing box in the wall of the steam chamber.

The weight flow in pounds per second has been already determined experimentally—equations (16) and (16*a*). The results coincide with what would be expected from a theoretical consideration of the matter. Thus—

$$w = \text{weight flow per sec.} = \mathrm{A}\, v\, \rho \text{ lb.} \quad . \quad (25c).$$

and v is given in equation (25*b*), while

$$\mathrm{P}^1{}_1 = \mathrm{K}\,\rho_1{}^n \text{ and } \mathrm{P} = \mathrm{K}\,\rho^n, \text{ hence } \rho = \rho_1\left(\frac{p^1}{p^1{}_1}\right)^{\frac{1}{n}}$$

and

$$w = \mathrm{A}\sqrt{\frac{2\,g\,n}{n-1} \times 144\, p^1{}_1\,\rho_1\left[\left(\frac{p^1}{p^1{}_1}\right)^{\frac{2}{n}} - \left(\frac{p^1}{p^1{}_1}\right)^{\frac{n+1}{n}}\right]} \quad . \quad (26$$

Now the same weight w lb. per second will pass any and every section of the nozzle; hence w is constant for every section. But $w = \mathrm{A}\, v\, \rho$; hence if A has a minimum value

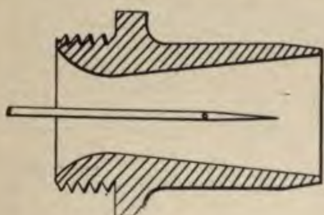


FIG. 70.

anywhere along the nozzle, the corresponding value of $v\,\rho$ must be a maximum, because w is constant all along the nozzle; hence the surd in equation (26) must be a maximum for the section in question, and therefore the quantity inside the bracket must be a maximum. Differentiating with respect to the variable p^1 and equating to zero we have

$$\frac{p}{p^1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \quad . \quad . \quad . \quad . \quad (27)$$

for the minimum section. Substituting in equation (26), or in (25*b*) and (25*c*), we have

$$v \text{ at minimum section } = \sqrt{2\,g \frac{n}{n+1}\, 144\, p^1{}_1\, V_1} \quad . \quad (28)$$

$$w \text{ lbs. per sec. } = A \left(\frac{2}{n+1}\right)^{\frac{1}{n-1}} \sqrt{2\,g \frac{n}{n+1}\, P_1\, \rho_1} \quad . \quad (29)$$

The value of w so found is the actual amount of discharge when A is the sectional area of the minimum section in square feet and the final or exit pressure of the steam is not greater than three-fifths of the initial pressure.

It is curious that the actual discharge w lb. per second should also be the maximum possible discharge; or, in other words, if the terminal pressure be anything below about three-fifths of the initial pressure, the nozzle always discharges the maximum possible amount of steam. This may be shown in the following manner:—

Let Q = the weight per second

$$= A\, v\, \rho = A_1\, v_1\, \rho_1 = A_2\, v_2\, \rho_2$$

substituting the value v from the equation

$$\frac{v^2}{2\,g} = \frac{n}{n-1}\left[\frac{P_1}{\rho_1} - \frac{P}{\rho}\right]$$

remembering $v_1 = 0$.

$$Q = A_2\, \rho \sqrt{2\,g \frac{n}{n-1}\left(\frac{P_1}{\rho_1} - \frac{P}{\rho}\right)} \quad . \quad . \quad (30)$$

This expression will be a maximum for a given value of

$$\frac{P_1}{\rho_1} \text{ when } \frac{P_1}{\rho_1}\rho^2 - \rho\, P$$

is a maximum. For P put $K\,\rho^n$, then the above expression becomes, after differentiating and equating to zero,

$$\frac{2\,P}{\rho_1}\rho - (n+1)\,\rho^n = 0$$

or

$$\frac{P_1}{\rho_1} = \left(\frac{n+1}{2}\right)\frac{P}{\rho} \quad . \quad . \quad . \quad . \quad (31)$$

Putting this value of $\frac{P}{\rho}$ in equation (30) we get

$$Q \text{ max.} = A \rho \sqrt{2 g \frac{n}{n-1} \left[\frac{P_1}{\rho_1} - \frac{2}{n+1} \cdot \frac{P_1}{\rho_1}\right]}$$

$$= A \rho \sqrt{2 g \frac{n P_1}{(n+1)\rho_1}}$$

$$= A \frac{\rho}{\rho_1} \sqrt{2 g \cdot \frac{n}{n+1} \cdot P_1 \rho_1} \quad . \quad . \quad . \quad (32)$$

From equation (31) we have—

$$\frac{\rho}{\rho_1} = \left(\frac{n+1}{2}\right) \frac{P}{P_1} = \frac{n+1}{2} \left(\frac{\rho}{\rho_1}\right)^n$$

Divide through by $\frac{\rho}{\rho_1}$, and we get

$$\frac{2}{n+1} = \left(\frac{\rho}{\rho_1}\right)^{n-1}$$

$$\text{or } \left(\frac{2}{n+1}\right)^{\frac{1}{n-1}} = \frac{\rho}{\rho_1} \quad . \quad . \quad . \quad . \quad (33)$$

Substituting this value in equation (32),

$$Q \text{ max.} = A \left(\frac{2}{n+1}\right)^{\frac{1}{n-1}} \sqrt{2g \frac{n}{n+1} \cdot P_1 \rho_1} \quad . \quad (34)$$

This last expression for the maximum weight flow of steam corresponds with what we find by experiment, and with that given by equation (29); hence the actual flow is always a maximum when the pressure at the down stream end of the nozzle is less than three-fifths of that at the up stream end.

Reverting to equation (31) for the relation between pressures and densities for maximum weight flow, we find that

$$\frac{\rho}{\rho_1} \cdot \frac{2}{n+1} = \frac{P}{P_1},$$

and putting in the value of $\frac{\rho}{\rho_1}$ from equation 33

$$\frac{P}{P_1} = \frac{2}{n+1} \cdot \left(\frac{2}{n+1}\right)^{\frac{1}{n-1}} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \quad . \quad . \quad (35)$$

Some values of $\frac{p^1}{p^1_1}$ will be found in the next table. It will be found by plotting that the following simple relation exists between the ratio $\frac{p^1}{p^1_1}$ and n, when p is the pressure at the throat, namely,

$$\frac{p^1}{p^1_1} = \cdot785 - \cdot182\, n \quad . \quad . \quad . \quad . \quad (36)$$

The different known values of . n.	$\frac{P}{P_1}$ in equations (27) and (35).	Co-efficient of $A\sqrt{P_1 \rho_1}$ in equation (34).	$\sqrt{2g\frac{n}{n+1}}$ co-efficient of $\sqrt{\frac{P_1}{\rho_1}}$ in equation (28).
1·4	·528	3·88	6·14
1·3	·546	3·79	6·04
1·135	·577	3·60	5·85
$\frac{10}{9}$	·582	3·57	5·82
1	·6	3·40	5·75

Some idea of the behaviour of the steam inside the nozzle can be obtained by the use of the exploring tube previously referred to (fig. 70]. With it an indicator diagram can be drawn, giving pressures corresponding to different points along the axis of the nozzle. Such a diagram is shown fig. 71.

The nozzle is shown at O N, and the direction of flow is indicated by the arrow. The pressure is plotted above the

base *o n*, giving the curves *b a p*, *b a q s*, and *b a q c*. The time during which the steam is passing through the nozzle is so small that the flow must be either adiabatic or nearly so, and hence we should expect to find that the velocity could be predicted by the application of the adiabatic condition to the usual equation of motion of a fluid. This is found to be the case except in so far as it may be influenced by friction and eddy motion in the nozzle. The terminal or exhaust pressure at the down stream end of

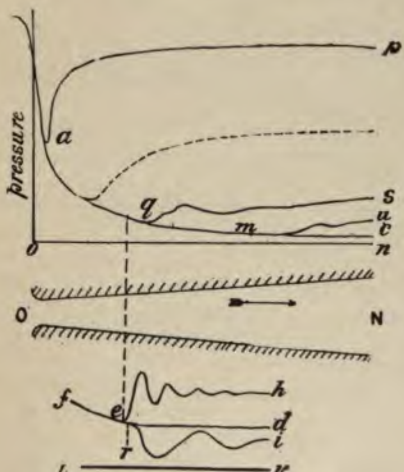


FIG. 71.

the nozzle appears to influence the character of the pressure curve near that end, but it does not affect the earlier part of the curve to any extent. Thus the curve *b a q c*, fig. 71, is obtained when the pressure at N is not much different from *c n*. If the terminal pressure is raised considerably, the later part of the curve becomes *q s*; while a very high terminal pressure gives the curve *a p*.

There seems to be a natural tendency for the pressure to follow the curve *b a q c*, and it appears that this is the

Boiler pressure per sq. inch $= \frac{P_1}{144}$.	Volume of 1 lb. of steam at a pressure P_1 lbs. per square foot. $= V_1$ cubic feet.	Product of the pressure per sq. foot into the volume of 1 lb. steam in cubic feet. $= P_1 V_1$ foot-pounds.	Density of the steam at a pressure of P_1 lbs. per square foot—*i.e.*, weight of 1 cubic foot steam. $= \rho_1$.	Temperature of steam in degrees Fah., corresponding to a pressure of P_1 lbs. per square foot. $= t_1^{\circ}$.	Sensible heat of 1 lb steam at temperature t_1°—*i.e.*, units of heat required to raise the temperature of 1 lb. of water from 32° Fah. to t°_1 Fah. $= h_1$.	Latent heat of 1 lb. of steam at a temperature t° Fah.—*i.e.*, units of heat required to convert 1 lb. of water at t_1° Fah. into 1 lb. of steam at t_1° Fah. $=$ L.	Square root of the product of pressure per square foot into the density $= \sqrt{P\rho}$
1	330·4	47577	·0030	102·0	70·04	1043·01	·6
2	171·9	49507	·0058	126·3	94·37	1026·09	1·3
3	117·3	50673	·0085	141·6	109·76	1015·38	1·9
4	89·51	51734	·0112	153·1	121·27	1007 87	2·5
5	72·56	52387	·0138	162·4	130·56	1000·90	3·1
10	37·83	54561	·0264	193 3	161·66	979·23	6·2
15	25·85	56097	·0386	213·0	181·60	965·32	9·2
20	19·74	56993	·0506	227·9	196 65	954·81	12·1
50	8·338	60565	·1199	280·9	250·35	917·26	29·4
70	6·076	61542	·1645	302·7	272·65	901·63	40·4
80	5·358	61719	·1866	311·8	281·95	895·11	44·8
90	4·796	62156	·2085	320·1	290·37	885·19	52·0
100	4·342	62737	·2302	327·6	298·09	783·77	97·0
110	3·969	62885	·2519	334·6	305·24	878·74	63·2
120	3·656	63177	·2735	341·0	311·88	874·07	69·0
130	3·390	63461	·2949	347·1	318·12	869·68	74·0
140	3·161	63705	·3163	352 8	324·00	865·55	80·0
150	2·962	63979	·3876	358·2	329·56	861·63	86·0
160	2·786	64180	·3588	363·3	324 85	857 91	91·0
170	2 681	64382	·4800	368·2	349·89	854·36	96·0
180	2·494	64620	·4012	372·8	344·70	850·96	102·2
190	2·368	64788	·4223	377·3	349·33	847-70	108 0
200	2·256	64525	·4483	381·6	358·76	844·57	112·1
250	1·825	65700	·5478	400·8	373·[illegible]0	830 45	140·4
300	1·585	66812	·6515	417·8	390·98	818·80	168·5
500	·942	67824	1·0717	467·4	448·50	781·02	278·5

curve which would be obtained if the nozzle delivered into a chamber in which the pressure was either zero or negligible. This corresponds to unrestricted expansion.

The reluctance of the curve to be influenced by terminal conditions is shown in the lower part of fig. 71, where the nozzle used terminated at *r*, the pressure being plotted above *t v*. The curve *f e* corresponds to *b a q* in the upper part of the figure, but to a larger vertical scale. If the pressure outside the nozzle were the same as that at the last point of the curve, the horizontal line *e d* was obtained; but, if lower, there was a sudden drop of pressure as the steam left the nozzle, while if the pressure were slightly higher, a sudden rise of pressure occurred beyond the nozzle, giving the curve *e h*.

In both cases where the pressure outside the nozzle was not the same as that of the natural curve *b a q c*, vibrations or stationary waves occurred, as indicated in the figure.

Some idea may now be gathered as to how a partial vacuum can be formed in an injector, enabling it to induce the feed water to flow up to it before starting.

The pressure curve between the beginning of the steam nozzle and the overflow chamber resembles *b a q m u*, the atmospheric pressure being *u n*. The steam will leave the nozzle at some such point as *m*, where the pressure is below that of the atmosphere. From *m* to *u* the pressure first falls slightly and then rises to that of the atmosphere as the kinetic energy is converted to pressure energy. The feed will easily flow into the region of lower pressure at *m*, and hence an injector with a properly designed steam cone can pick up its feed water prior to starting, provided the channel through which the steam passes after leaving the steam cone is sufficiently large to prevent choking.

CHAPTER VI.

HIGH-PRESSURE COMPOUND INJECTORS.

LET β_2 be the ratio of water to steam in the forcer, while β_1 represents the same ratio in the lifter; then for every pound of steam that issues from the "forcer" steam nozzle there are $(1 + \beta_2)$ lb. of water and steam that enter the boiler. Of the β_2 lb. of water in the forcer, we shall have $\left(\frac{1}{1 + \beta_1}\right)$th part steam (condensed) and the remainder cold feed water; *i.e.*, $\frac{\beta_2}{1 + \beta_1}$ lb. condensed steam and $\frac{\beta_1 \beta_2}{1 + \beta_1}$ lb. of feed water. Therefore the total amount of steam used in both forcer and lifter per pound of steam in the forcer

$$= 1 + \frac{\beta_2}{1 + \beta_1} = \frac{1 + \beta_1 + \beta_2}{1 + \beta_1} \text{ lb.}$$

Net amount of feed water per pound of steam in the forcer

$$= \frac{\beta_1 \beta_2}{1 + \beta_1} \text{ lb.}$$

Therefore the ratio,

$$\frac{\text{Net cold feed water}}{\text{Total steam}} = \frac{\beta_1 \beta_2}{1 + \beta_1 + \beta_2} = \beta, \text{ say.}$$

Let v_8 be the final velocity of the delivery jet; then, if the steam for both forcer and lifter is taken from the boiler, we shall have the total momentum of the two steam jets equal to the momentum of the final delivery jet, or

$$v_2 = v_8 (1 + \beta),$$

and $$v_8 = \frac{v_2}{1 + \frac{\beta_1 . \beta_2}{1 + \beta_1 + \beta_2}} = \frac{(v_2 (1 + \beta_1 + \beta_2)}{(1 + \beta_1)(1 + \beta_2)}$$

This equation is only an approximation, as the pressure P_8 in the delivery jet has been neglected. From experiments on compound injectors, this has been found to be small.

We may find the temperature t_8 of the delivery jet thus—

$$1 + \beta_1 = \frac{1114 + \cdot 3\, t_1 - t_5}{t - t_5} \quad . \quad . \quad (37)$$

also

$$1 + \beta_2 = \frac{1114 + \cdot 3\, t_1 - t_3}{t_8 - t_3}$$

therefore

$$t_8 = t_5 + \frac{1114 + \cdot 3\, t_1 - t_5}{1 + \beta_1} + \frac{1114 + \cdot 3\, t_1 + t_3}{1 + \beta_2}$$

$$= \frac{t_5 (\beta_1 \beta_2 + \beta_2 - 1) + (1114 + \cdot 3\, t_1)(1 + 2\,\beta_1)}{(1 + \beta_1)(1 + \beta_2)}$$

The compound injector is most advantageous when the "lifter" portion is an exhaust steam injector. The feed water is thereby heated to the same temperature as it would be with an ordinary single exhaust injector, and at the same time the "forcer" drives it into the boiler at any steam pressure now in general use, while the temperature is still further raised by the aid of the live steam.

The principal feature of the compound injector is the high temperature to which it will raise the feed water; so that in calculations with regard to this apparatus we may take approximate values for the velocity of the exhaust steam and of the delivery jet.

If we make $t_1 = 212$ deg. in equation (37), we have—

$$1 + \beta_1 = \frac{1178 - t_5}{t_8 - t_5}$$

and

$$t_8 = t_5 + \frac{1178 - t_5}{1 + \beta_1} + \frac{(1114 + \cdot 3\, t_1)(1 + \beta_1) - \beta_1\, t_5 - 1178}{(1 + \beta_1)(1 + \beta_2)}$$

The gain of heat by the feed water due to the exhaust injector may be found thus: There is 1 lb. of exhaust steam for every β_1 lb. of feed water. This pound of steam contains $(1178 - t_5)$ thermal units, reckoned above

t_5 deg., all of which would have been wasted, but by the aid of the injector are returned to the boiler. This, then, is evidently the net gain *(or saving)* of heat per β_1 lb. of cold feed water, and hence the saving per pound of water is

$$\frac{1178 - t_5}{\beta_1} \text{ thermal units.}$$

Each of these pounds of water would require

$$1082 + \cdot 3t^1 - (t_5 - 32) \text{ thermal units}$$

to evaporate it in the boiler; therefore the saving in heat expended per cent

$$= \frac{(1178 - t_5)\,100}{(1114 + \cdot 3\,t_1 - t_5)\,\beta_1} \quad . \quad . \quad . \quad (38)$$

In the same manner, every β_1 lb. of cold feed water have mixed with them 1 lb. of exhaust steam; hence out of $(1 + \beta_1)$ lb. of feed that enter the boiler, the save in water must be 1 lb., or

$$\frac{100}{1 + \beta_1} \text{ per cent} \quad . \quad . \quad . \quad . \quad . \quad (39)$$

We see from this that it is expedient to use as little feed water as possible, or, in other words, as much exhaust steam as possible. If we assume that $t_5 = 60$ deg., ${}_1t =$ 368 deg., corresponding to a pressure of 170 lb. per square inch absolute, and $\beta_1 = 7{\cdot}5$, corresponding to a temperature of $t_3 = 192$ deg., we have from (38) and (39) the save in heat $= 12{\cdot}75$ per cent, and the saving in feed water $= 11{\cdot}75$ per cent.

CHAPTER VII.

Exhaust Injectors.

From the nature of the supply of exhaust steam from a direct-acting engine, it is evident that the *re-starting* capabilities of an exhaust instrument must be of the most sensitive description. This feature is the very essence of successful working, and without which the exhaust apparatus remains an inert piece of material only.

A longitudinal section of an injector made by Messrs. Davies and Metcalf, and also by Messrs. Schäffer and Budenberg, is shown (fig. 72) as fitted to feed a stationary boiler. In appearance it resembles the flap-nozzle re-starting injector, with the exception that the steam nozzle is very large, the combining cone of extra length, and a conical spike is fixed axially in the steam cone. The excessive-size of the steam orifice has already been predicted in the mathematical passages of the previous

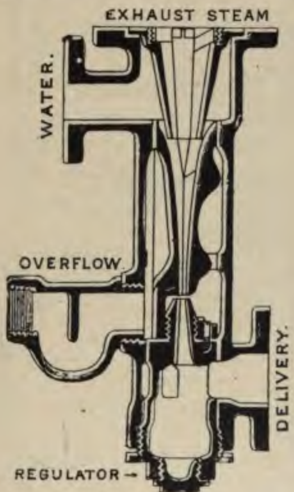


FIG 72

pages. The spike, it is said, tends to assist in the concentrating of the jet of exhaust steam. It will be noticed that the overflow pipe is of peculiar construction, having a vertical wall across the main portion; at the same time the end of the pipe remote from the injector dips below the surface of the water in a tank. The object is to prevent the inlet of air to the combining cone of the injector, which would immediately stop its action. This may be gathered from the previous mathematical treat-

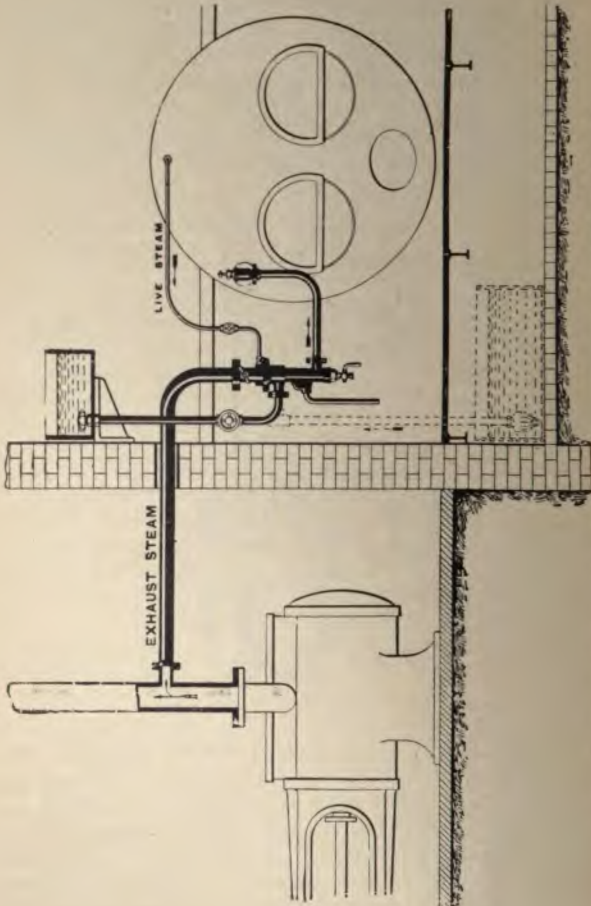


FIG. 73.—Messrs. Holden and Brooke's General Arrangement of an Exhaust Injector.

ment, where it was shown that the pressure in the combining cone may be considerably less than that of the atmosphere.

The arrangement of an exhaust injector is shown in fig. 73. The junction with the exhaust pipe is perfectly plain, and free from projections, which tend to the increase of back pressure in the cylinder, while the water must be supplied to the apparatus from above the injector. A small throttle valve, having a hand-wheel attached, is placed in the steam pipe just above the injector, to regulate the supply of exhaust steam. It has been previously mentioned that to secure the maximum efficiency of any injector the steam must be as dry as possible. The steam coming from the exhaust port of an engine often contains a considerable and variable amount of moisture, none of which should enter the injector. To secure this end the junction with the exhaust pipe should never be made in such a position that condensed steam can gravitate to the injector. Examples of general practice may be gathered from the various diagrams which follow. The above remark regarding the junction with the exhaust pipe should not be lightly passed over, as, if neglected, the efficiency of the apparatus may be impaired.

The supply of water coming into the combining cone can be regulated by turning the nut at the delivery end (fig. 72), thus making the combining cone advance towards or recede from the steam cone. It must be remembered that, should the injector be placed with its axis horizontal, the flap must *always open upwards* for best working. It can be made to open in a horizontal direction, and it has worked when opening downwards. If it is necessary to work the injector when the engine is standing, a small pipe connects the steam space in the boiler with the steam space in the injector just below the wheel valve. In this way the boiler steam may be wire-drawn down to the pressure of the atmosphere, and used exactly as the exhaust steam from the engine. Should the boiler pressure be between 70 lb. and 105 lb. per square inch, the makers recommend the use of the supplementary live steam nozzle (fig. 74), whereby the jet is assisted by live steam from

the boiler *after* it has been mixed with the exhaust steam. The figure explains itself.

The exhaust injector of Messrs. Holden and Brooke is of a somewhat different type to the one just described, but works within about the same limiting conditions.

A longitudinal section is shown in fig. 75. The combining cone has four transverse slits for the purpose of

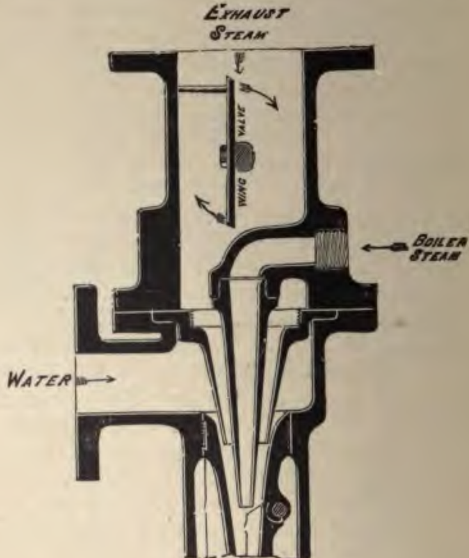


FIG. 74.

assisting it at starting in the same manner as the corresponding slits behave in the high-pressure injector. For forcing against a pressure up to 80 or 90 lb. per square inch this injector is used as it is shown in fig. 75, but for pressures up to 120 lb. per square inch the supplementary supply of boiler steam is used as shown in fig. 76; while for pressures up to 200 lb. per square inch the compound principle is used as shown in fig. 77.

When it is desired to work the injector fig. 75 while the engine is at rest, a pipe connects the chamber at X with the boiler, through which steam is admitted after closing W.

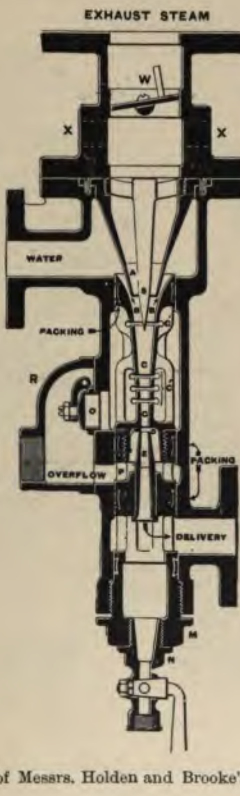


FIG. 75.—Section of Messrs. Holden and Brooke's Exhaust Injector.

The exhaust (left hand) portion is their ordinary form of exhaust injector, which they use for all purposes when the pressure of the boiler to be fed does not exceed 75 lb. or 80 lb. per square inch. It much resembles the same

firm's re-starting influx live-steam injector, and differs from it in having three extra transverse slits in the combining cone near the overflow orifice; the four slits communicating with a separate chamber, whose outlet is covered

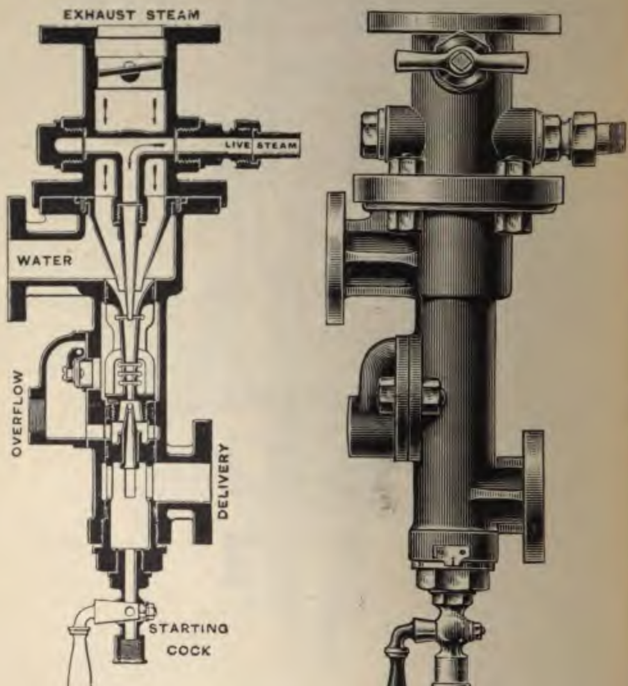


FIG. 76.—Messrs. Holden and Brooke's Exhaust Injector.

by an ordinary flap valve. These slits and flap valve act in the same manner as the flap nozzle just previously described, and render the action perfectly automatic. The amount of cold feed water can be regulated by turning the nut at the bottom of the delivery chamber, the annular

er orifice being increased or decreased thereby, as in flap-nozzle instrument. The combining cone is ngthened and maintained in true shape by longitudinal , shown in the figure in elevation. The chamber ounding the flap valve is bolted to the main casing, can be removed without affecting the remainder of

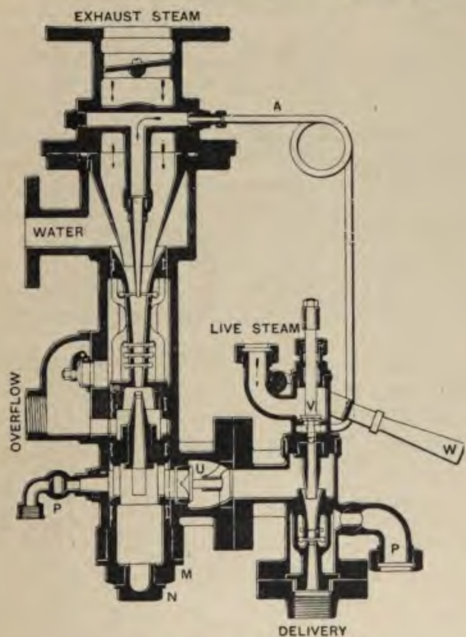


Fig. 77.

apparatus. The supplementary or right-hand portion olted to the delivery flange of the exhaust injector. onsists of an ordinary simple live-steam injector, with overflow (P) capable of being closed by means of the er (W) which opens the live-steam supply valve.

s soon as the lever is moved steam is conveyed, by ns of the tube A, to the wing valve, and is admitted

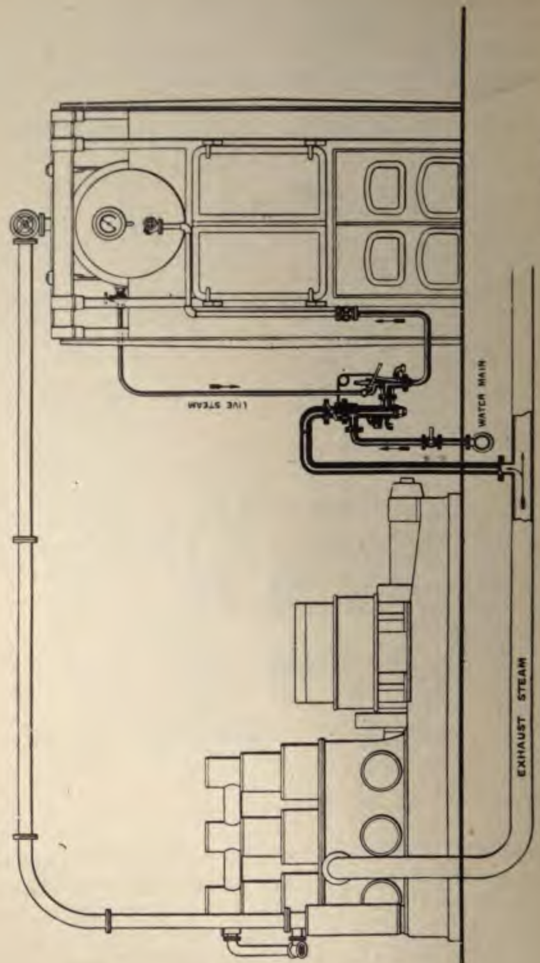


FIG. 78.—General arrangement of Messrs. Holden and Brooke's Compound Exhaust Injector.

to the hollow spindle of the exhaust injector, forming a supplementary live-steam jet, which aids the action of the exhaust portion. Water is now delivered to the supplementary injector through the non-return valve (U), which prevents live steam entering the exhaust injector if it should throw off while working. On further raising the lever live steam is admitted to the cones of the supplementary injector, and at the same time the overflow is closed, when the water is forced into the boiler at a very high temperature.

The method of fitting up the compound exhaust injector is shown in fig. 78.

These injectors are also used on locomotives, the exhaust steam being taken from the root of the blast pipe and conveyed to the injector by a pipe containing a grease separator F, fig. 79, to prevent the cylinder oil from getting into the boiler. In the early days this was the most serious drawback to the exhaust injector, and unless the grease is properly removed from the exhaust steam, much more harm may be done to the boiler than can be balanced by the saving in fuel obtained.

The special exhaust injector for locomotive work made by Messrs. Davies and Metcalf is shown in section in fig. 80.

The exhaust steam enters at the left-hand end, and the cold feed water at the lowest flange. The portion A is the exhaust injector, and B the supplementary portion. A non-return valve separates the two portions.

A casting crowns the overflow flange. This contains an overflow valve, which is maintained upon its seating, while the injector is at work, by the pressure of the delivery jet being communicated to a little piston on the top of the valve spindle. The necessity for this is evident with a compound apparatus. There is a small pipe seating A close to the water regulator, which is connected by a small pipe to another pipe seating A on the live steam supply pipe. By the use of this pipe the injector may be worked when the engine is standing. The overflow pipe has a non-return valve in it, so that the atmosphere and its pressure may be kept out of the combining tube.

The construction of the supplementary portion or

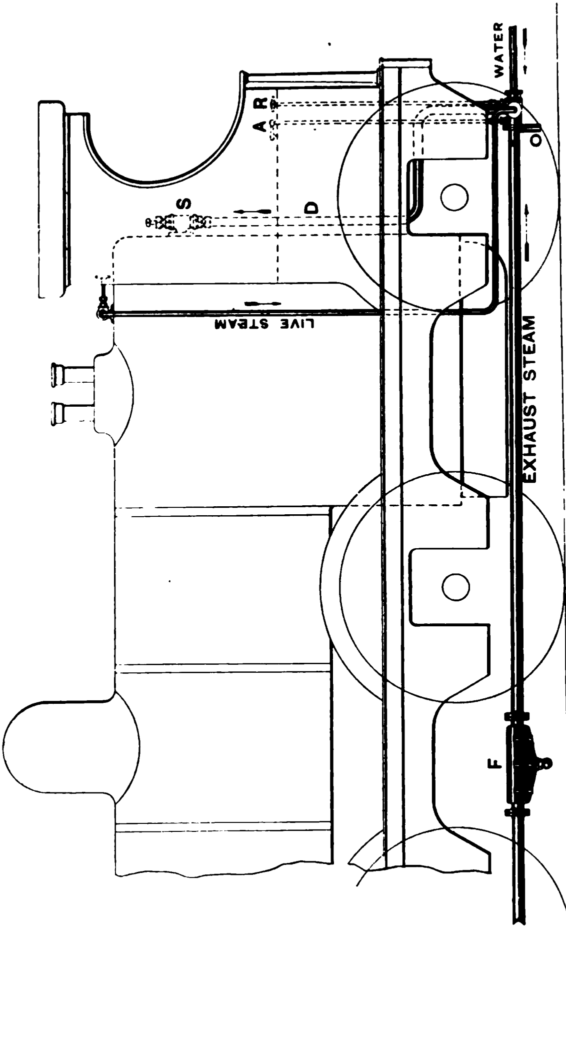


FIG. 79.

"forcer" and the supplementary overflow valve is shown in the section, fig. 80. When the injector is starting, the

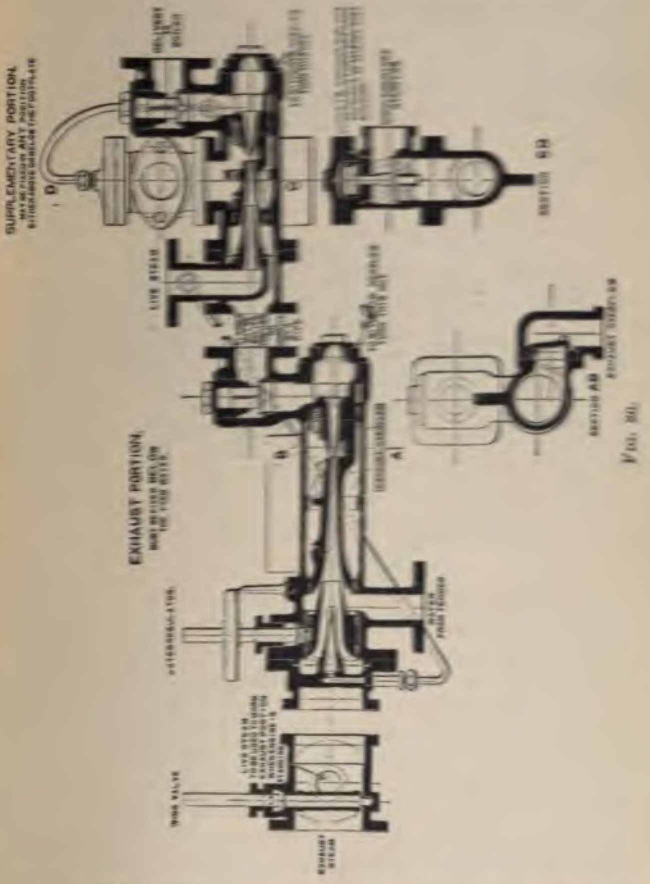
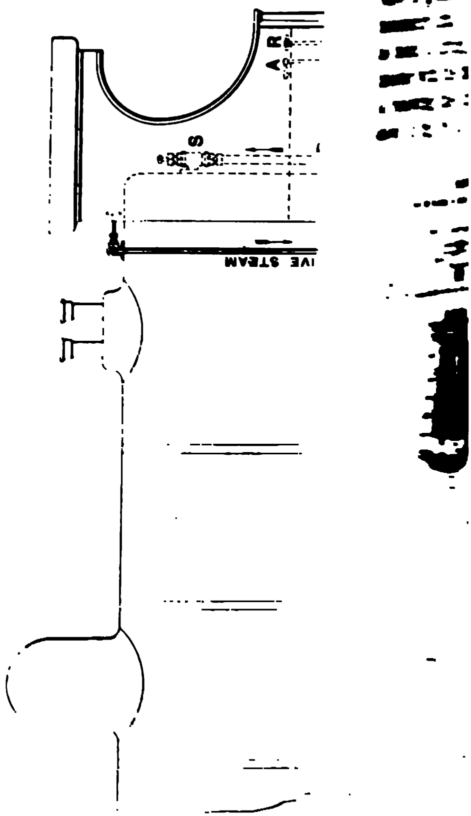


Fig. 80.

pressure above the little piston of this valve, which is protected by a rubber diaphragm, is the same per square



…e is induced to flow to the in…

…ase or deposit from the cyl…
…and thus reduce its evapo…
…endanger the life of the firebox…
…comes from the blast pipe, …
…box, containing a perforated cy…
…towelling, fig. 92, and is cal…

GREASE SEPARATOR.

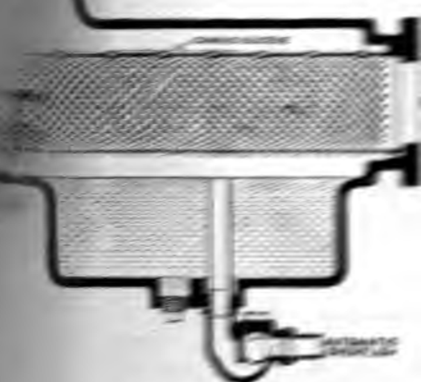


FIG. 92.

…parator." This is the invention of Mr. M… …ted on all locomotives with his firm's ej… … It must be remembered that it is only a… …motives, where a strong blast is necessary f… …uction of a good draught.

…ese injectors, which are fitted on one side only… …en great satisfaction to [illegible] drivers. The boilers… …more freely than with [illegible] ordinary single line…

inch as that below the valve itself, namely, that of the steam jet in the combining cone; and as the area of the valve is greater than the area of the piston, the valve necessarily lifts. Thus the supplementary portion is quite as much a lifting and re-starting apparatus as a single injector with the flap nozzle. In order to avoid the trouble of renewing the indiarubber, the equivalent arrangement shown in fig. 81 is adopted. In this it will be noticed that

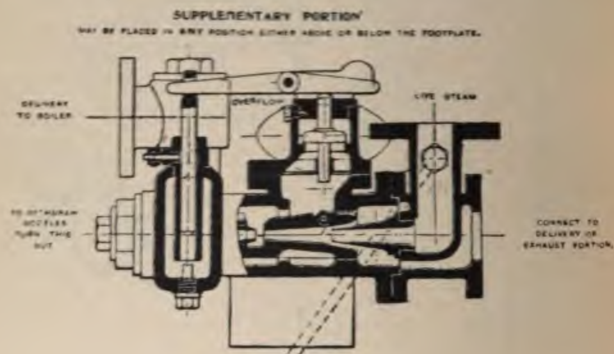


Fig. 81.

a lever is introduced, against one end of which the overflow valve spindle presses, whilst the other end of the lever rests upon a small cylindrical rod, the lower end of which acts as a piston, transmitting the pressure by means of the lever to the overflow valve, instead of the small pipe shown in fig. 80. This arrangement (Metcalfe Bros.' patent) is practically indestructible, and removes all possible trouble, due to the possible failure of the diaphragm, without warning, as indiarubber is a very treacherous material under the best circumstances. To start this injector, the water is turned on first, and then the live steam to the supplementary portion, after which the injector immediately starts to work. By turning on the live steam, the air in the exhaust portion is exhausted, and the exhaust

steam from the blast pipe is induced to flow to the injector instantly.

To prevent any grease or deposit from the cylinders entering the boiler, and thus reduce its evaporative efficiency, as well as endanger the life of the firebox, the exhaust steam, as it comes from the blast pipe, passes through a rectangular box, containing a perforated cylinder clothed with Turkish towelling, fig. 82, and is called a

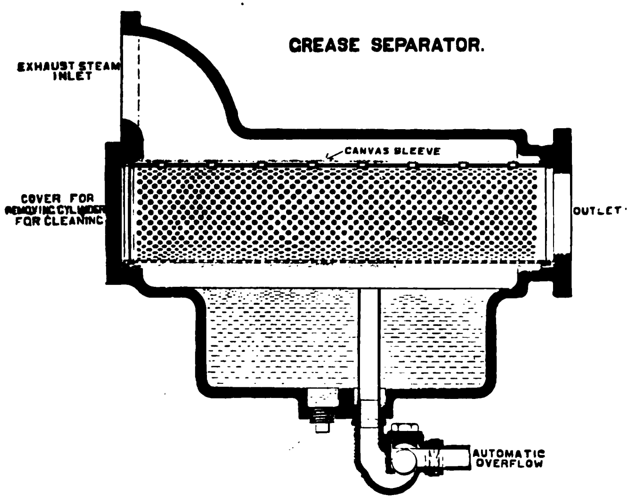


FIG. 82.

"grease separator." This is the invention of Mr. Metcalfe, and is used on all locomotives with his firm's exhaust injectors. It must be remembered that it is only used on locomotives, where a strong blast is necessary for the production of a good draught.

These injectors, which are fitted on one side only, have given great satisfaction to the drivers. The boilers steam much more freely than with the ordinary single live-steam apparatus. The average saving in coal consumption over similar sister engines, working the same traffic, is about

4 lb. of coal per train mile, or about 10 per cent. After the drivers got over their prejudice, they preferred engines fitted with the exhaust apparatus, and found no more difficulty in using them than those worked by live steam only.

The amount of water delivered per hour is approximately the same as that delivered by a similarly-sized single exhaust injector; but the temperature of the delivery is raised about 60 deg. or 70 deg. above that with an exhaust injector alone. Generally speaking, the temperature of delivery of the exhaust portion ranges between 180 deg. and 200 deg., and the temperature of the feed as it enters the boiler about 260 deg. It is thus evident that a considerable saving of coal and water ought to ensure from the use of exhaust steam in the "lifter," and at the same time the feed water enters the boiler at a temperature of from 80 deg. to 100 deg. higher than when fed by a single live-steam injector, and the stresses in the boiler shell, set up by the cooling effect of the comparatively cold feed water, are avoided to a great extent. These injectors take feed water up to 80 deg. Fah., in which case the supply tank should be above the combining cone. For hot climates a special injector is designed to take feed up to 90 deg.

A few remarks with respect to the method of working and maintaining these injectors may not be out of place here. The live steam supply to the supplementary portion should always be maintained at full boiler pressure, and never wire-drawn; hence the steam stop valve must always be left full open. The supply of exhaust steam may vary under different conditions of load; hence the exhaust steam wing valve may want altering in position under various circumstances. The exhaust steam should not exceed the atmospheric pressure when it reaches the injector; hence when working entirely with live steam, that which goes to the exhaust portion must be wire-drawn considerably. The condition of the steam in the injector may be judged from the appearance of the overflow.

Under ordinary circumstances no running at the overflow will take place; but should there be grit or other

obstruction in the injector, or should the temperature of the feed water be too high before it approaches the injector, then running at the overflow may be expected.

Through the sediment deposited in the cones of injectors working in districts where the water is bad the cones have to be periodically removed, and the scale cleaned off the internal surfaces. Much the same thing happens, though less in degree, when bad oil is used for lubricating the cylinders. To facilitate the operation of removing the scale, Messrs. Holden and Brooke supply cleaning bits of the required shape and size. These consist of a piece of flat mild steel, $\frac{1}{8}$ in. thick, cut in profile to the exact shape of the cone to be cleaned, and in appearance like an ordinary template. The cone should be removed from the injector, and, if necessary, soaked in petroleum, to soften the scale before using the cleaning bits. Care must be exercised not to remove any metal from the cones when using the bits. If required, the cone may be further cleaned by turning a piece of soft wood to the same shape as the bit; but on no account is any grinding material, such as emery powder, to be used with it.

The makers recommend that the water inlet to the exhaust portion should be at least 2 ft. below the lowest level of the feed-water tank. When erecting the injector fittings the red lead should be applied to the joints with care so that dried portions shall not break off and find their way into the cones. An asbestos packing for joints is preferable on this account. After erection or alteration the cones are generally removed, and water blown through the pipes to clear them of all substances that may have got in during erection.

The length of period which an injector will go without cleaning depends entirely upon the quality of the water and steam used. There have been many cases in which years have elapsed without removing the cones. It seems somewhat strange that it is possible for little bits of sediment to stick to the surface of an injector cone when moving at the rate of 100 ft. per second, or more. In the author's opinion, it is not while the injector is at work that deposit accumulates, but at every stoppage a certain

amount of water flows through the cones and leaves them covered with a film of water, which, of course, contains sediment in solution. The cones being at a temperature very near the boiling point, this film of water

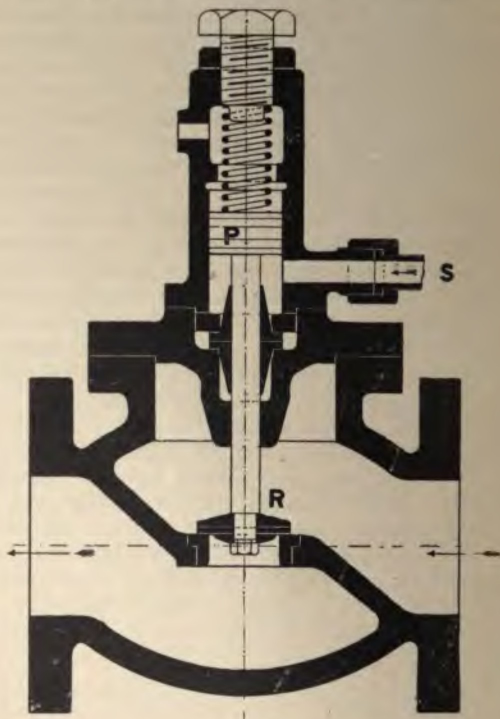


FIG. 83.

is quickly evaporated, and leaves behind a certain amount of deposit.

In the case of engines working intermittently, such as winding engines at a colliery, the continual stopping and starting would necessitate the same number of openings

and closings of the water stop valve, so that water would not be wasted. This difficulty is surmounted by the application of Johnson's automatic control, as supplied with Messrs. Holden and Brooke's injectors when fitted to the above class of engines. It is shown in section in fig. 83, and is really a stop valve fitted to the cold water supply pipe. The valve is opened by the pressure of steam on the under side of the small piston P, the steam pipe S being connected to the steam chest of the engine. Directly steam is turned on to work the engine, it flows down the pipe S and opens the water valve R, maintaining it open until the steam is shut off from the engine, when the spiral spring maintains the valve on its seating. In this way the exhaust injector is perfectly independent of the attendant, and no water is wasted. These injectors have been fitted to numerous colliery engines, and continue to work well without any trouble.

Quantity of Feed per Hour with Exhaust Injector.

With any exhaust injector we may easily find approximately the amount of feed water supplied to the boiler per hour. Let Q = this quantity in pounds, then $\frac{Q}{3600}$ lb. are delivered per second; and if D = the diameter of the delivery cone orifice in inches, its area must be

$$\frac{\pi}{4} \times \frac{D^2}{144} \text{ square feet,}$$

and the amount of steam and water fed to the boiler per second = area × velocity × density,

$$= \frac{\pi}{4} \times \frac{D^2}{144} \times v_3 \times \rho$$

Substitute v_3 by its value given in the equation

$$v_3 = \frac{v_2}{1 + \beta}$$

and assume that v_2 = 1,200 ft. per second; also that β

$= 7$ (this is nearly its minimum value). We then find that the total delivery per second

$$= \frac{\pi}{4} \times \frac{D_2}{144} \times 62{\cdot}5 \times \frac{1200}{8} = 51\ D^2.$$

Of this quantity of water and steam delivered every 1 + β) lb. are made up of β lb. of water and 1 lb. condensed steam; hence of the above quantity $\frac{\beta}{1+\beta} \times 51\ D^2$ lb. are feed water, and $\frac{1}{1+\beta} \times 51\ D^2$ lb. are condensed steam. Then

$$\frac{7}{1+7} \times 51\ D^2 = \frac{Q}{3600}$$

or

$$Q = 160650\ D^2$$

where D is measured in inches.

If D should be measured in millimetres, then

$$Q = 160650 \times \frac{D^2}{25{\cdot}4^2}.$$

The weight of steam used per hour in pounds

$$= \frac{Q}{\beta} = \frac{Q}{7}\ \text{lb.}$$

Volume of steam used per minute

$$= \frac{Q}{7} \times \frac{\text{volume of 1 lb. at 212 deg.}}{60}$$

$$= \frac{Q \times 26{\cdot}37}{420} = {\cdot}063.\quad Q\ \text{cubic feet.}$$

This being the volume of steam used by the injector, the supply from the exhaust must never be less than this amount. So that the injector shall not have its action impaired for want of steam, makers recommend that for good working there should be at least one and one-third times this amount, and when an early cut-off happens,

such as with a locomotive or a variable expansion engine, the minimum volume of steam should be twice that given by the above equation.

The rule given by Messrs. Holden and Brooke to determine the maximum size of injector that can be used is this:

Find the volume swept out by the piston per minute in cubic feet, and multiply it by the fraction three-quarters; then look in the subjoined table, and see what number of injector will require this amount. If the cut-off takes place very early, replace the fraction three-quarters by one-half.

Delivery Table for Exhaust Injectors.

Size of injector.	2½	3	3½	4	4½	5	6	7
Minimum cubic feet of steam required per minute	43	60	80	100	126	156	225	306
Maximum of water delivered in gallons per hour	95	150	200	270	340	420	600	830

Size of injector.	8	9	10	11	12	13	14	20
Minimum cubic feet of steam required per minute......	400	506	625	756	900	1056	1225	2500
Maximum of water delivered in gallons per hour	1080	1370	1700	2050	2450	2870	3330	6800

The above gives the maximum amount of feed delivered by the minimum amount of steam. The data which really must determine the size of the injector is the size of the boiler to be fed, and an exhaust injector should not be any larger than is necessary; otherwise the boiler will get filled too quickly, and the working must then be intermittent, and thus the maximum saving by the use of the exhaust steam will not be obtained. This is a point which should receive careful consideration when procuring an injector. It may appear strange that the above figures are given almost irrespective of the pressure of the steam when first admitted into the cylinder of the engine; but upon a little

per pound of steam, and v_a the velocity of efflux of the exhaust steam. Then, from the equality of momenta, we have—

$$v_2 = \frac{v_a}{1 + \beta}$$

Now, from equation (25) we have—

$$v_a = v_2 = \sqrt{2\,g\,\frac{n}{n-1}\left(\frac{P_a}{\rho_a} - \frac{P_2}{\rho_2}\right)}$$

Also we have $\qquad P = K\,\rho^n$

therefore $\qquad \left(\frac{P}{K}\right)^{\frac{1}{n}} = \rho$

and $\qquad \frac{P}{\rho} = \frac{P}{\left(\frac{P}{K}\right)^{\frac{1}{n}}} = K^{\frac{1}{n}}\,P^{\frac{n-1}{n}}$

Substituting in (25) with suffixes, we find

$$v_a = \sqrt{2\,g\,\frac{n}{n-1}\,K^{\frac{1}{n}}\left(P_a^{\frac{n-1}{n}} - P_2^{\frac{n-1}{n}}\right)} \quad . \quad (A)$$

The value of v_a thus determined depends to a very great extent upon the value given to n. That value depends upon the quality of the steam; and in the exhaust pipe of an engine it probably varies considerably under different circumstances. Thus, in fixing a value for v_a we must allow a margin for variation of circumstances. Should the exhaust steam be wet, as it often is, then the dry steam has to do more work than it would have if the whole of the steam were dry; and therefore the velocity of the exhaust steam will be reduced in proportion to its wetness. We may obtain an approximation to the velocity v_a by assuming the steam dry and $n = \frac{10}{9}$ in equation (A), where the most convenient form for calculation is—

$$\log v_a = 3{\cdot}6106579 + \frac{1}{2}\log\left(2{\cdot}15499 - 144^{\frac{1}{10}}\,p_2^{\frac{1}{10}}\right)$$

In this we have assumed the pressure of the atmosphere 15 lb. per square inch.

The author has calculated several different values of v_a, corresponding to different values of P_3, from which the curve number 1, fig. 85, has been plotted.

Now to obtain the amount of feed water per pound of steam that will give the combined jet its maximum velocity. For steady motion equation (3) must hold. Neglecting v_4, and the loss of head h, the velocity with which the jet enters the boiler, we have

$$\frac{P_4 - P_3}{\rho} = \frac{v_2^2}{2g}$$

and by replacing the right-hand side by its equivalent we have

$$2g\,\frac{(P_4 - P_3)}{\rho} = \left(\frac{v_a}{1+\beta}\right)^2$$

For v_a we may substitute its value in (A), and the above then becomes

$$2g\,\frac{P_4 - P_3}{\rho} = \frac{n}{n-1}\,\frac{K^{\frac{1}{n}}}{(1+\beta)^2}\left[P_a^{\frac{n-1}{n}} - P_3^{\frac{n-1}{n}}\right]$$

in which we have three variables, viz., $(P_4 - P_3)$, $(1+\beta)$, and P_3. It will now be more convenient to replace the two last by their equivalents in terms of the temperature t_3.

Rankine has given us the following accurate relation between temperature and pressure:

$$\log_{10} P = A - \frac{B}{T} - \frac{C}{T^2} \quad . \quad . \quad . \quad . \quad . \quad (B)$$

in which P = pressure in pounds per square foot, $A = 8{\cdot}2591$, T = the absolute temperature Fah. of the vapour (saturated steam), $B = 2731{\cdot}6$, and $C = 396{,}945$; $\log_{10} B = 3{\cdot}43642$, and $\log_{10} C = 5{\cdot}59873$. (*Vide* "The Steam Engine and other Prime Movers," by Rankine, p. 237.)

Multiply both sides of equation (B) by $\mu = 2{\cdot}30258$, the modulus for converting common to Napierian logarithms, and we have

$$\log_e P = \mu \left[A - \frac{B}{T} - \frac{C}{T^2} \right]$$

and, after reversing the operation of taking logarithms, we find that

$$P = e^{\mu \left[A - \frac{B}{T} - \frac{C}{T^2} \right]}$$

and, consequently,

$$P^{\frac{n-1}{n}} = e^{\mu \left(\frac{n-1}{n}\right) \left(A - \frac{B}{T} - \frac{C}{T^2}\right)}$$

After substituting in above

$$2g \frac{P_4 - P_3}{\rho} = \frac{n K^{\frac{1}{n}}}{n - 1} \left(\frac{T_3 - T_5}{1178 - t_5}\right)^2$$

$$\left[e^{\mu \left(\frac{n-1}{n}\right) \left(A - \frac{B}{Ta} - \frac{C}{Ta^2}\right)} - e^{\mu \left(\frac{n-1}{n}\right) \left(A - \frac{B}{T_3} - \frac{C}{T_3^2}\right)} \right]$$

To find what value of T_3 will make this expression a maximum, differentiate the right-hand side with respect to T_3, and equate the first differential co-efficient to zero. The temperature T_5 is constant. We then find that—

$$\mu \left(\frac{n-1}{2n}\right) \left(T_3 - T_5\right) \left(\frac{B}{T_3^2} + \frac{2C}{T_3^3}\right) e^{\mu \left(\frac{n-1}{n}\right) \left(A - \frac{B}{T_3} - \frac{C}{T_3^2}\right)}$$

$$= \left[e^{\mu \left(\frac{n-1}{n}\right) \left(A - \frac{B}{Ta} - \frac{C}{Ta^2}\right)} = e^{\mu \left(\frac{n-1}{n}\right) \left(A - \frac{B}{T_3} - \frac{C}{T_3^2}\right)} \right]$$

Divide both sides of this equation by the latter term of the right-hand side, and we have—

$$\mu \left(\frac{n-1}{2n}\right) (T_3 - T_5) \left(\frac{B}{T_3^2} + \frac{2C}{T_3^3}\right)$$

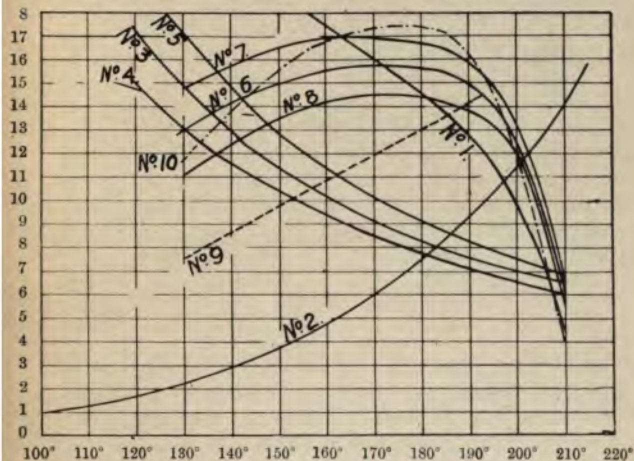
$$= \frac{e^{\mu \left(\frac{n-1}{n}\right) \left(A - \frac{B}{Ta} - \frac{C}{Ta^2}\right)}}{e^{\mu \left(\frac{n-1}{n}\right) \left(A - \frac{B}{T_3} - \frac{C}{T_3^2}\right)}} - 1$$

$$= e^{\mu \left(\frac{n-1}{n}\right) \left[B \left(\frac{1}{T_3} - \frac{1}{Ta}\right) + C \left(\frac{1}{T_3^2} - \frac{1}{Ta^2}\right) \right]} - 1.$$

But

$$e^x = 1 + x + \frac{x^2}{1 \times 2} + \frac{x^3}{1 \times 2 \times 3} \times \text{etc., } ad\ infinitum,$$

and if x is a small fraction, all terms beyond the second can be neglected with respect to those before it. The exponent of e in the above equation cannot be greater than ·0909,



Temperatures of mixed water and steam in combining cone.

FIG. 85.

Maximum pressures in tens of pounds per square inch, against which the injector will feed.
Velocities in hundreds of feet per second of the exhaust steam.
Pounds of feed water per pound of steam.
Pressures per square inch in combining cone.

1.—t_3, v_a.
2.—t_3, p_3.
3.—t_3, β, $t_5 = 60°$.
4.—t_3, β, $t_5 = 50°$.
5.—t_3, β, $t_5 = 70°$.
6.—t_3, v_3, $t_5 = 60°$.
7.—t_3, v_3, $t_5 = 50°$.
8.—t_3, v_3, $t_5 = 70°$.
9.—t_3, v_3; max. flow, t_5, $= 60°$.
10.—t_3, p_4, $t_5 = 60°$.

and therefore in the expansion we can safely neglect all terms after the second, without introducing an error of

more than one-third of one per cent. The above then becomes—

$$\frac{(T_3 - T_5)}{2}\left(\frac{B}{T_3^2} + \frac{2C}{T_3^3}\right) = B\left(\frac{1}{T_3} - \frac{1}{T_a}\right) + C\left(\frac{1}{T_3^2} - \frac{1}{T_a^2}\right)$$

If we assume that $T_5 = 520$ deg.—*i.e.*, $t_5 = 60$ deg. Fah.—and put in the values of B and C given above, together with $t\,a = 212$ deg. Fah., the last equation reduces to—

$$\cdot 00363\, T_3^3 - T_3^2 - 1110\, T_3 + 150000 = 0,$$

a solution of which is found when T_3 is put equal to 648 deg. Fah. *absolute*, which is equivalent to about 187 deg. Fah.

This we may then take as the temperature of the water as it enters the boiler (when supplied to the injector at 60 deg.) when moving with its maximum velocity. The corresponding amount of feed water per pound of steam is given by the equation—

$$1 + \beta = \frac{1178 - t_5}{t_3 - t_5}$$

from which we find that $\beta = 7{\cdot}9$ approximately.

Now, the velocity of the steam in the steam cone given by the curve No. 1, fig. 85, when the temperature in the delivery cone is 185 deg., is 1,370 ft. per second. Putting this in equation (54*a*), we have—

$$v_3 = \frac{v_a}{1 + \beta} = \frac{1370}{8{\cdot}9} = 154 \text{ ft. per second.}$$

If there were no resistances in the injector or its connections (especially the latter), this would force water against the pressure P_4 given by equation—

$$\frac{P_4 - P_3}{\rho} = \frac{v_3^2}{2g}.$$

From the curve No. 2, fig. 85, showing the relation between temperature and pressure, we may find P_3. We then find that $p_4 = 171$ lb. per square inch. The delivery water often takes a circuitous route from the injector to

the boiler, passing through chambers and orifices, and by valves, whose construction and form are of the very opposite to that which the principles of hydraulics would indicate; and hence we must expect a loss of head between the delivery cone and the boiler. This loss of head sometimes amounts to as much as 60 per cent of the total head. The pressure against which the exhaust injector will generally feed in practice is about 70 lb. per square inch; though with a little manipulation it is possible to reach a higher pressure, as the following results of actual tests will show:—

t_5	p_4	t_3	t_3-t_5	β	No. of Injector.	
54	84	156	102	10	4	
81	55	182	101	9·8	4	
50	65	180	130	7·7	4	This injector would not start when t_5 was greater than 75.
60	165	..	—	—	10	A little water issued from the overflow.

The above results are collected from a paper read before the Owens College Engineering Society, by Mr. George Harrison, the injectors used being made by the Patent Exhaust Injector Company, Manchester.

In the fourth experiment the whole of the quantities were not taken, but supposing we assume that the quantity of feed water was such as to give a maximum velocity to the delivery jet, we should find that $\beta = 7{\cdot}9$ approximately, and the temperature of the jet within 20 deg. of the ordinary boiling point.

The chief causes which tend to make the injector feed against lower pressures than those indicated by calculation are radiation, fluid friction, eddies, wetness of exhaust steam, and the resistance of chambers whose forms change abruptly, all of which have been neglected in the calculation. It may be roughly estimated that an injector will feed against a pressure of about 60 per cent of that indicated by calculation when the resistances are neglected therefrom.

diverging or delivery cone, we may make the pressure in the delivery pipe much greater than either the pressure in the combining cone or that of the source of the steam.

If now we increase the amount of cold water to three or four times the former amount per pound of steam, the temperature of the mixture must necessarily be much less than in the previous case, and consequently the pressure of the vapour must be much less, say 4 lb. per square inch. But on account of the steam having to convey the increased amount of water, the pressure against which it will deliver that water must be considerably less than in the former case.

It so happens that the pressure against which this increased amount of cold water can be delivered is slightly greater than that of the atmosphere; while if we connect the steam orifice of this modified instrument to the exhaust pipe of an engine, the trifling pressure in the combining cone, together with the rapid condensation of the exhaust steam as it comes in contact with the cold water, allows of a fairly good vacuum in the cylinders and exhaust pipe. This, of course, materially adds to the amount of work that can be obtained from an engine; the modified injector performing the functions of condenser and air pump combined.

In this modified apparatus it is necessary to present as much cold water surface to the steam as possible, and at the same time to remove as much resistance to the passage of the water as is convenient in construction, so that the work done by the exhaust steam will be a minimum. To attain this end the steam and cold water orifices change places, and the water flows through the condenser under a small head, or the condenser itself is situated below the surface of the cold-water supply tank.

If we require the amount of condensing water per pound of steam, we merely have to insert the different values for temperatures in the equation

$$\beta = \frac{1114 + \cdot 3\, t_1 - t_3}{t_3 - t_5}$$

assuming the steam to be dry at exhaust. If the dryness fraction is known, then

$$\beta = \frac{h_1 + x_1 L_1 - h_3}{t_3 - t_5}$$

When the ejector condenser is supplied with condensing water with a head, then the water cone is so shaped that the flow of water tends to produce a vacuum, in which case the water does not need the assistance of the steam in forcing back the atmosphere; but should the ejector be placed in the water—*e.g.*, in a tank or canal—then the energy of the steam must assist in driving the water through the apparatus. In this latter case the supply of water is greatly in excess of that required for condensing the exhaust steam. Assuming that the ejector receives no assistance from atmospheric pressure, or from a head of water, then if v = velocity of steam in the steam cone, and v_3 = the velocity of the combined jet after mixture, we have—

$$\frac{v}{1 + \beta} = v_3.$$

If the combined jet is delivered at atmospheric pressure, through a trumpet mouth-pipe, we must have, neglecting the final velocity—

$$\frac{P_2}{\rho} + \frac{v_3^2}{2g} = \frac{P_4}{\rho};$$

where P_4 is the pressure of the atmosphere per square foot. The pressure p_3 will range between zero and about 4 lb. per square inch; on the average, say 2 lb. per square inch. If we neglect P_3, we shall then have a minimum value of v_3, given by—

$$\frac{v_3^2}{2g} = \frac{15 \times 144}{62 \cdot 5};$$

or v_3 must not be less than about 47 ft. per second. The maximum quantity of condensing water is then given by—

$$\frac{v}{47} - 1 = \beta \max.$$

The velocity v may be assumed to be 1,300 ft. per second, and we then find that β max. $= 27$ approximately. When the condenser is fed from a higher level, the quantity β may be much greater than 27.

The general arrangement of the ejector condenser invented by Mr. Alexander Morton, of Glasgow, is shown (fig. 86) in end and side elevation. The exhaust pipe E leads directly into the top casting of the ejector T. The condensing water enters by the pipe A, while the condensed steam and water leave the ejector by the pipe C. The

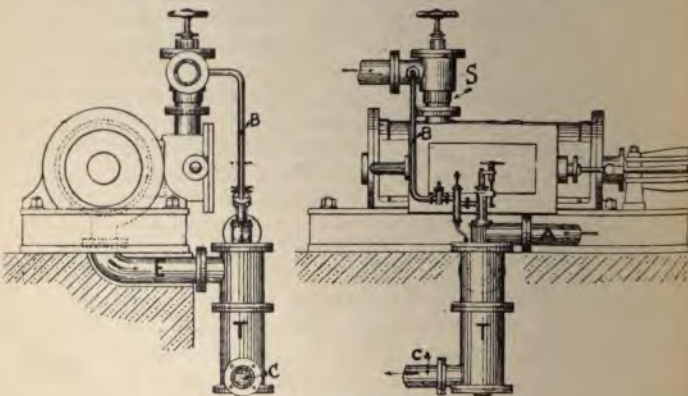


FIG. 86.

main steam pipe from the boiler to the engine is shown at S, and a small supplementary live-steam pipe B, the object of which will be explained further on.

A section of the simple non-lifting condenser is shown, fig. 87. The channel B delivers the water into the throat A, where it comes into contact with the steam, which approaches the water through the channel D. The surface of water exposed to the steam is that of the jet exposed between A A and F F. The delivery cone G is so shaped that the velocity of the water and condensed steam at exit is small. The space surrounding the cone G acts as an air chamber. The rod C C, tapered at its lower end, can be

raised or lowered by the hand wheel and screw, and thus the amount of condensing water can be regulated to the

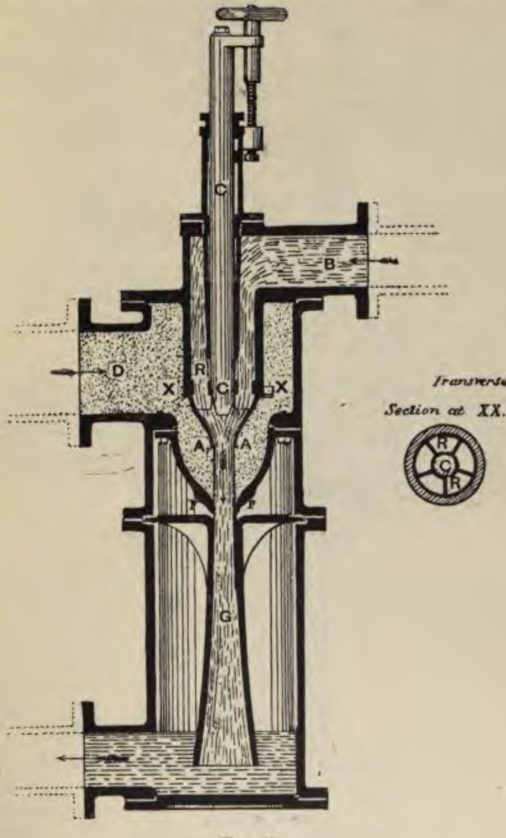


FIG. 87.

amount of steam. The five vanes at R act partly as a guide for the spindle C, and partly to maintain the direction of the water parallel to the axis of the ejector, so that any

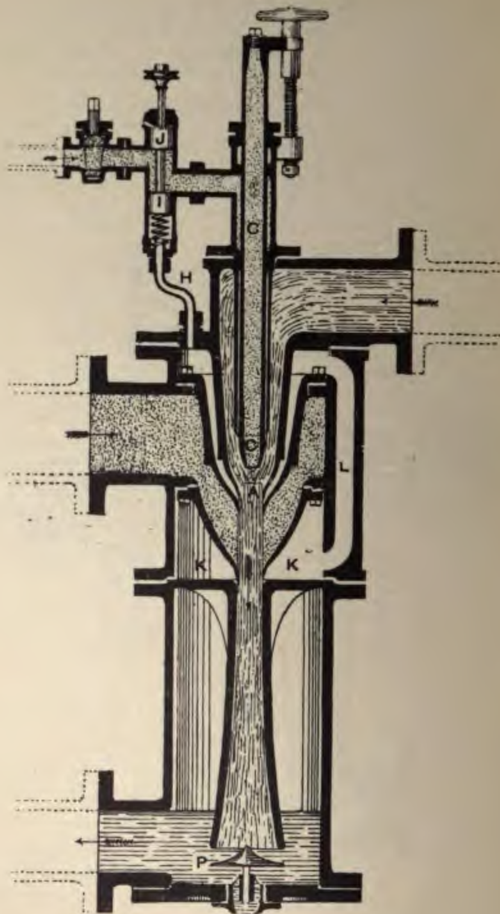


FIG. 88.

tendency towards vortex motion of the stream lines will be checked.

A special form of this ejector is given in fig. 88, in which the water has to be lifted to the exhaust steam cone by means of a small steam jet. The rod C C is made hollow, and perforated near its top end. A small pipe allows steam to flow direct from the main steam pipe into the hollow rod C C, past the two little pistons J, I. The upper side of the piston J is in communication with the atmosphere, and the lower side of I connected to the chamber L K by the small pipe H. If the amount of condensing water is insufficient, there will be a poor vacuum at K, and the pressure in that chamber will assist the spiral spring in forcing up the piston I, so that a free passage of live steam may pass to the ejector. When the steam jet is drawing up too much water, the vacuum at K will allow the atmosphere to force

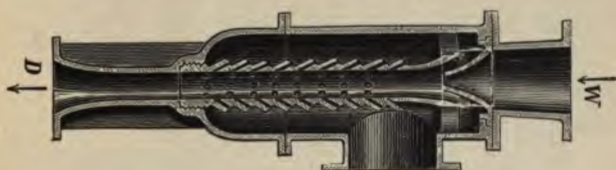


FIG. 89.

down the piston J, and close, or partially close, the passage for the live steam. In this way the amount of live steam is automatically regulated to produce the best working vacuum. In other respects this apparatus is identically the same as the one just described.

Another form of ejector condenser is shown in fig. 89, and is manufactured by Messrs. Körting Bros. The exhaust steam enters the condensing cone, and mixes with the water by means of a series of inclined channels drilled in the combining cone, the water having a straight path through the ejector. A non-return valve is generally placed somewhere between the cylinder and the condenser, to prevent any water from being syphoned back into the cylinder.

The makers recommend this fixed capacity ejector in all cases where a head of 15 ft. or more of condensing water

is available. The velocity attained by the water is then sufficient to produce a vacuum without the assistance of the exhaust steam, and therefore the condenser is independent of the load on the engine, or the amount of steam used,

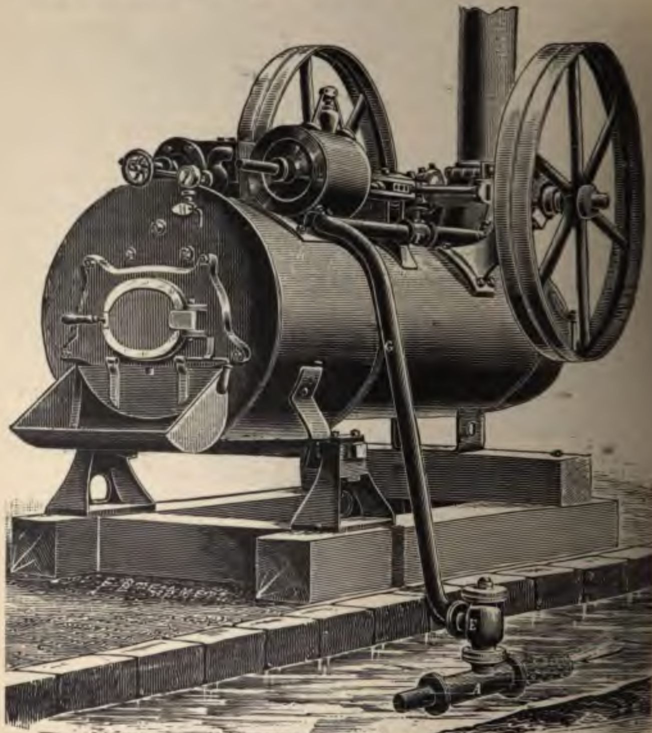


FIG. 90.

and will maintain a steady vacuum if it is large enough to take the maximum quantity of steam.

Should the available head of water be less than 15 ft., and the variation of the amount of exhaust steam be small, an injector of slightly different proportions to the one just

described may be used, and the injector situated beneath the surface of the water. Such an arrangement is shown

Fig. 91.

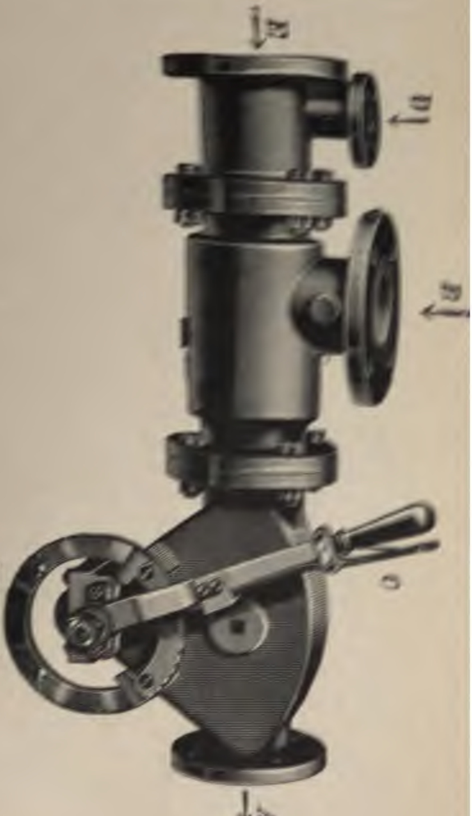


Fig. 92.

in fig. 90, the body of the ejector being at A, the non-return valve at E, the exhaust pipe at G, and the stop valve at F. The flow of water in the river or channel will

remove all the hot water and supply fresh cold water to the condenser. Should it be situated in a pond of still water, then the delivery pipe must be made longer, to carry away the hot water.

It may happen that the load upon the engine may vary from time to time, and the supply of condensing water derived from a source which will not allow of the head of 15 ft. previously spoken of. In this case the adjustable capacity condenser should be used, a section and elevation of which are shown in figs. 91 and 92. The construction is something similar to fig. 89, but the delivery tube is made capable of sliding axially, and thus allowing of any desirable number of steam passages being open at the same time. The motion on the tube D, fig. 91, is derived from a connection with the handle C, fig. 92. At D we have a small live-steam pipe for the purpose of charging the condenser when it is situated above the water supply. It may also be used to induce a flow of condensing water while the engine is at work, should it be necessary. The exhaust steam enters at B, while the water comes into the condenser at E. The minimum capacity with this happens when the tube D is moved as far as possible towards E, and *vice versa*. The variation of capacity ranges between 10 and 100 per cent. These condensers can be fixed either vertically or horizontally, but the vertical position is to be preferred.

A few results of experiments with an adjustable capacity are given on page 147. The water was supplied under favourable conditions, with a head varying from 1 ft. to 6 ft. The same results also happen with any head under 15 ft. The temperature of the water was 50 deg. Fah.

On page 147 the position of the lever termed "closed" is that in which the tube D covers up all the small steam channels, and leaves only the top aperture next to the water cone for the inlet of exhaust steam. There are nine sets of small steam channels, corresponding to the nine notches or positions of the lever. In the above experiments the maximum amount of steam was coming from the engine, and hence the best vacuum was obtained with the condenser full open.

It has also been found that when the lever is full open

the condenser will work with so little water as 15 lb. per pound of steam, and give a vacuum of from 20 in. to 24 in. Less than that proportion, the vacuum rapidly falls off, and the condenser will soon stop.

Ratio of water to steam.	Position of lever.	Vacuum obtained in inches of mercury.
200	closed	23·5
140	1st notch	26·5
125	2nd notch	27·0
110	3rd notch	27·5
100	4th notch	28·0
90	5th notch	28·0
80	6th notch	28·0
70	7th notch	28·0
60	8th notch	28·0
50	9th notch	28·25

The result of some experiments with a fixed capacity condenser are given below. The head of water was

Ratio of water to steam.	Vacuum in inches of mercury.
160 to 80	26·75
70 to 41	25·50
30 to 25	23·50
20 to 18	21·50
17 to 16	20·00
13 to 12	16·00
11 to 10	12·00
10 to 9	8·00
9 to 8	4·00

The above figures represent maximum efficiency.

15 ft., and the temperature of condensing water from 50 deg. to 55 deg. Fah.

CHAPTER X.

The Water Injector.

With an ordinary steam injector the motive power is obtained from steam under a given pressure; this pressure being equivalent to a certain head (or height of a column) of steam. If we replaced the column of steam by a column of water we should still have the same phenomenon

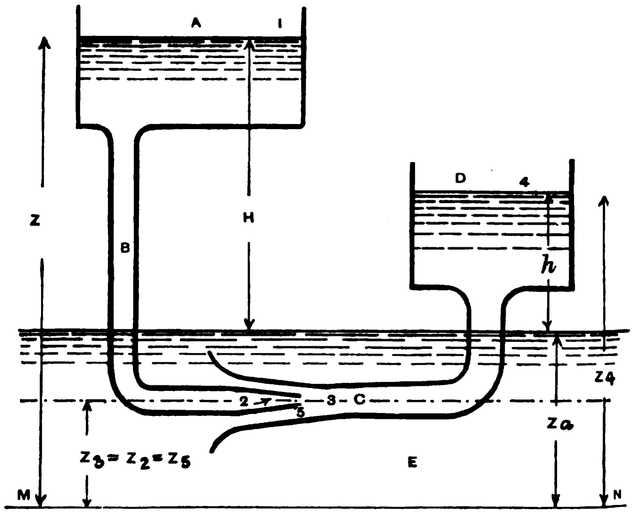


Fig. 93.

produced, namely, the motive power fluid (water) taking up water and delivering it against a certain head; but the amount of head against which the water will be forced will be less than when steam was used as the motive fluid, and the amount of water taken up will, of course, be much less.

In fig. 93 the tank A contains the motive power fluid, and the reservoir E holds the water, which has to be

pumped up into the tank D. Water descends from the tank A by the pipe B, and is delivered into the trumpet-mouthed tube C, which is in connection with the tank D. Let M N be any datum line, and assume the surface of the water in the tanks A and D to be z_1 and z_4 respectively above the datum line, while the axis of the injector is z_3 above it. Also let the surface of the water in the reservoir be at a distance z_a from the datum line, while v_1, v_2, v_3, v_4, and v_5 are the velocities of the water—(1) at the surface of the tank A, (2) at the orifice of the pipe B, leading into the injector, (3) at the narrowest part of the pipe C, (4) at the surface of tank D, and (5) at the annular orifice between the pipes B and C.

For the motion of the water between (1) and (2), we must have for steady motion

$$\frac{P_1}{\rho} + \frac{v_1^2}{2g} + z_1 = \frac{P_2}{\rho} + \frac{v_2^2}{2g} + z_2.$$

Now, v_1 can be neglected, and P_1 is that of the atmosphere P_a, the pressure P_2 is the same as P_3, while $z_2 = z_3$, and hence the above becomes

$$\frac{v_2^2}{2g} = \frac{P_a - P_3}{\rho} - z_3 + z_1.$$

Again, between (3) and (4) we must have

$$\frac{P_3}{\rho} + \frac{v_3^2}{2g} + z_3 = \frac{P_4}{\rho} + \frac{v_4^2}{2g} + z_4.$$

We may neglect v_4, while $P_4 = P_a$, and thus obtain

$$\frac{v_3^2}{2g} = \frac{P_a - P_3}{\rho} - z_3 + z_4.$$

Also for the bell-mouthed portion of the tube C and the reservoir surface, we have

$$\frac{P_a}{\rho} + \frac{v_a^2}{2g} + z_a = \frac{P_5}{\rho} + \frac{v_5^2}{2g} + z_5,$$

of which v_a is zero, $P_5 = P_3$, and $z_5 = z_3$. After substituting these values, we find that

$$\frac{v_5^2}{2g} = \frac{P_a - P_3}{p} - z_3 + z_a.$$

If we represent the right-hand side of this equation by x, while we assume the height of the surfaces (1) and (4) above the reservoir to be H and h respectively, and substitute these quantities in the above equations, we get

$$v_2 = \sqrt{2g(H + x)}$$
$$v_3 = \sqrt{2g(h + x)}$$
$$v = \sqrt{2gx}$$

Now, from the laws of impact we have

$$(1 \times v_2) + (\beta \times v_5 = (1 + \beta) v_3,$$

when 1 lb. of motive-power water from the tank A takes up β pounds of water from the reservoir and delivers it into the tank D. Replacing the several velocities by their equivalent in (77), we have—

$$\beta = \frac{\sqrt{H + x} - \sqrt{h + x}}{\sqrt{h + x} - \sqrt{x}} \quad . \quad . \quad . \quad . \quad (D)$$

As x represents a perfect square, it cannot be negative. With positive values of x, we find that β increases with an increase of x. Writing down the equation—

$$x = \frac{P_a - P_3}{\rho} - z_3 + z_a$$

we see at once that as P_a is constant, x can be increased by either making P_3 less, or z_3 less, or by increasing z_a. The most convenient way of increasing x is the first method, and this may be done by suitably shaping the trumpet-mouthed tube C.

In the above apparatus the work done by the motive power $= H - h$ foot-pounds per pound of water coming out of the tank A; at the same time this pound of water takes

up β lb. of water from the reservoir and raises it through h ft. into the tank D, doing βh foot-pounds of useful work. The efficiency of the apparatus must then be

$$\frac{\beta h}{\mathrm{H} - h} = \frac{h}{\mathrm{H} - h} \cdot \frac{\sqrt{\mathrm{H} + x} - \sqrt{h + x}}{\sqrt{h + x} - \sqrt{x}},$$

which can be written as—

$$\frac{(\sqrt{h + x} - \sqrt{x})(\sqrt{h + x} + \sqrt{x})}{(\sqrt{\mathrm{H} + x} - \sqrt{h + x})(\sqrt{\mathrm{H} + x} + \sqrt{h + x})}$$

$$\times \frac{\sqrt{\mathrm{H} + x} - \sqrt{h + x}}{\sqrt{h + x} - \sqrt{x}}$$

$$= \frac{\sqrt{h + x} + \sqrt{x}}{\sqrt{\mathrm{H} + x} + \sqrt{h + x}} \quad . \quad . \quad . \quad . \quad (\mathrm{E})$$

This quantity also increases as x increases. As an example, take $\mathrm{H} = 150$ ft. and $h = 40$ ft., then from (D), when x is put equal to the successive values 0, 4, 8, 12, 20, the corresponding values of β are ·94, 1·24, 1·34, 1·46, and 1·62. Also from (E), when x equals 0, 4, and 8 successively, the efficiencies are ·34, ·45, and ·505, respectively.

The diameters of the orifices may be easily obtained, as we know the several velocities. For instance, let it be $\mathrm{H} = 150$, $h = 40$, and $x = 8$. Using the same subscripts as in the previous argument, we have the water raised per second

$$= \frac{1000 \times 10}{3600} \text{ pounds}$$

for every pound of water used from the tank A, there are β lb. pumped from the reservoir, and hence $1 + \beta$ lb. that pass through the pipe C. Therefore, the weight of water passing through C per pound of water pumped will be

$$\frac{1 + \beta}{\beta} \text{ pounds}$$

and weight flowing through C per second

$$= \frac{1000 \times 10}{3600} \times \frac{1 + \beta}{\beta} \text{lb.}$$

$$= \frac{100}{36} \times \frac{2 \cdot 34}{1 \cdot 34} \text{lb.}$$

the value of β being previously found from above. The volume of the above in cubic feet must be

$$\frac{100 \times 2 \cdot 34}{36 \times 1 \cdot 34} \times \frac{1}{62 \cdot 5}$$

and the diameter of the orifice (3) is given by

$$\frac{100 \times 2 \cdot 34}{36 \times 1 \cdot 34 \times 62 \cdot 5} = \frac{\pi}{4} d_3^2 \times v_3,$$

or, $$d_3 = \cdot 042 \text{ ft.} = \cdot 51 \text{ in.}$$

Again, the diameter of the orifice (2) may be obtained thus—

$$\frac{1000 \times 10}{3602} \times \frac{1}{\beta} \times \frac{1}{62 \cdot 5} = \frac{\pi}{4} d_2^2 \times v_2,$$

hence $$d_2 = \cdot 0203 \text{ ft.} = \cdot 245 \text{ in.}$$

The annular orifice (5) must have an area given by

$$\frac{1000 \times 10}{3600} \times \frac{1}{62 \cdot 5} A_5 \times v_5,$$

or, $$A_5 = \cdot 0057 \text{ square feet} = \cdot 735 \text{ square inches.}$$

These water injectors are not often used except in isolated cases where a great head of water is at hand. When dock gates have to be examined or removed, the water that has leaked through into the bulkhead compartments must be pumped out. This may be easily accomplished by attaching a flexible pipe to the hydraulic mains and sinking the water injector below the surface of the water to be pumped out. Of course the injector may be situated above the surface of the water if necessary.

The injectors used for this purpose at the docks at Barry are supplied with water under a pressure of 700 lb. per square inch, and deliver about 5,000 gallons per hour. The maximum lift of the injector is about 10 ft. and the average height above the injector through which the water is forced is 35 ft. These injectors answer their purpose well and require little attention.

CHAPTER XI.

AIR INJECTORS.

THE same reasoning may be applied to the pumping of air or other gases by the action of steam jet.

Let β lb. of air be pumped by 1 lb. of steam, the velocity of the steam being V, that of the mixture being v, and that of the air as it approaches the steam jet being $V_{,,}$, then we have the equation of momentum—

$$(1 + V) + \beta V_{,,} = (1 + \beta) v,$$

or,

$$v = \frac{V + \beta V_{,,}}{1 + \beta}.$$

The equivalent head of steam is $\frac{V^2}{2g}$, that of the delivered air $\frac{v^2}{2g}$. The efficiency of the apparatus as a pump must then be

$$\frac{\beta \frac{v^2}{2g}}{\frac{V^2}{2g}} = \frac{\beta\left(1 + \frac{V_{,,}}{V}\beta\right)^2}{(1 + \beta)^2}$$

This shows that the efficiency increases with the velocity $V_{,}$ of the incoming air. This, of course, will increase with the diminution of the pressure in front of the steam jet orifice, or, in other words, the increase of the vacuum there.

These steam-jet air pumps may be used for two distinct purposes, namely, (1) for forcing or inducing a current of air through chambers of pipes, such as for ventilating or producing a blast, or (2) they may be set to take the air out

FIG. 94.

of a given vessel, and so produce a partial vacuum there. The construction of an example of one of the first of these is shown in fig. 94, the top portion being in section. The double conical trunk contains a series of cones, the first of

which is supplied with steam from the small pipe on the left. The small hand wheel is merely for regulating the supply of steam. The use of a number of cones will be readily understood from what immediately follows.

With an ordinary steam-water injector the delivery jet moves down the diverging cone with considerable velocity, generally of more than 100 ft. per second. We could use this jet of water as the motive power jet of a water injector, and thus pump an additional quantity of water, but, of course, against a less head than when the extra cones were not used.

The head against which air is forced in blowing a fire is at the most only a few inches of water, and sometimes only

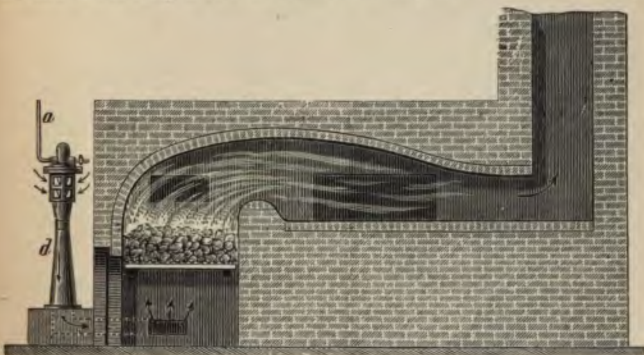


FIG. 95.

a fraction of an inch, and it is conducive to economy to force as much air as possible with the least amount of steam, and hence the use of a number of cones to augment the supply of air. The general arrangement of one of these steam-jet blowers, as applied to a heating furnace, is shown, fig. 95, *a* being the small steam pipe; the direction of the arrows denote the course of the air current. The illustrations have been kindly supplied by the makers, Messrs. Körting Brothers. These blowers are used as a means to produce forced draught, and will, with a pressure of steam of 45 lb., produce a blast of about 2 in. of water. Forced

draught enables more fuel to be burnt in a given time than with the ordinary draught, while by its aid the worst of coal and combustible material may be used in the fire.

The amount of air delivered per minute, together with the relative sizes of the apparatus, may be gathered from the subjoined table:—

Purpose of apparatus.	Air delivered per minute in cubic feet.	Diameter of steam nozzle in inches.	Minimum diameter of	
			Steam pipe in inches.	Air conduit in inches.
Undergrate blower....	130	$\frac{1}{12}$	$\frac{1}{2}$	
	400	$\frac{1}{7}$	$\frac{3}{4}$	
	550	$\frac{1}{16}$	1	9
	800	$\frac{1}{5}$	$1\frac{1}{4}$	10
	1,200	$\frac{1}{4}$	$1\frac{1}{4}$	12
	1,600	$\frac{2}{7}$	$1\frac{1}{2}$	13
	2,400	$\frac{1}{3}$	$1\frac{1}{2}$	16
	3,200	$\frac{3}{8}$	$1\frac{1}{2}$	18
Ventilator	1,000	$\frac{1}{8}$	$\frac{3}{4}$	14
	2 000	$\frac{3}{16}$	$\frac{3}{4}$	21
	4,000	$\frac{1}{4}$	1	30
	8,000	$\frac{5}{16}$	$1\frac{1}{4}$	40
	12,000	$\frac{3}{8}$	$1\frac{1}{2}$	48
	20,000	$\frac{1}{2}$	$1\frac{3}{4}$	60

CHAPTER XII.

AIR EJECTORS.

WHEN the steam jet is used for producing a partial vacuum, it is designed to remove as much air as possible out of a single vessel into which there is no leakage, rather than the carrying of a large quantity of air some distance. An example of this may be cited in the ejector used to exhaust the chambers of centrifugal pumps of air, and so charge them with water at starting. Another instance is the ejector used so universally for producing a partial vacuum in the cylinders and reservoirs of railway brakes. It is for this purpose it has received its widest application, and the brake itself would be of little use were it not for this appliance. A vertical section of the ejector used on the engines of the Great Western Railway is given (fig. 96). The main casting is fixed to the back outside firebox plate, the pipe A passing right through the boiler. Another hole, B, in the boiler plate conveys steam to the channel C, and to the back of the valve D, the latter being operated by the handle H. A port in the valve, for a certain position of the handle) directly coincides with the conical aperture E, whose axis is parallel to the axis of the pipe A. When the valve is turned into this position, steam is allowed to flow through the ejector, and thus exhaust the train pipe F and reservoirs of most of the contained air. The flap valve G prevents a return of the atmosphere into the train pipe when the steam is not passing through the ejector. The brake is applied by a movement of the handle H, whereby ports in the plate K are uncovered and a free passage is provided for the atmosphere into the chamber L, and thence, by a side port (the end of which is shown at M), into the train pipe. The vacuum brake on this railway is generally applied to the train only, the engine being fitted with a steam brake, which is applied and released by the handle (H) at the same time that the vacuum brake is undergoing the same operation. This is accomplished by

casting another port, which is controlled by the valve D, which admits boiler steam to the steam brake cylinder.

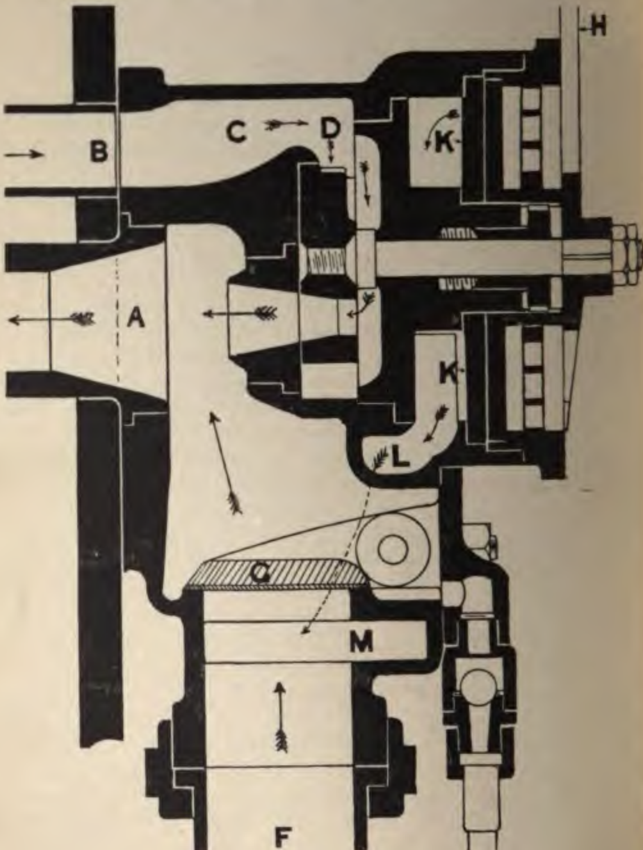


FIG. 96.

Another example of an ejector for the same purpose is shown in section (fig. 97), and is made by Messrs. Gresham

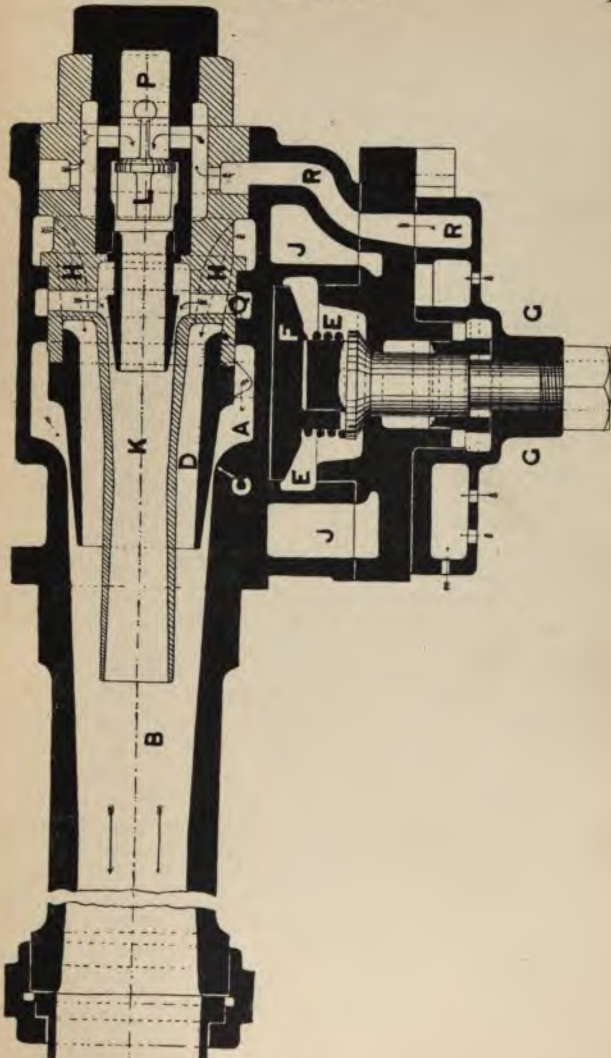


Fig. 97.

and Craven, for use with their vacuum automatic brake. This ejector exhausts the train pipe and brake apparatus at starting, and also maintains the vacuum during the time the train is running by ejecting any air which may leak into the train pipe. The first of these operations is performed by admitting steam into the cavity A from which it issues into the exhaust pipe B by the small annular orifice C. The diminution of pressure in the exhaust pipe, just beyond the orifice, causes a flow of air from the region of high-pressure D to the origin of low-pressure B, the air being carried along with the steam into the atmosphere through the pipe B. The space D is in communication with the main train pipe, a non-return valve being interposed to prevent the return of the air when the ejector is not at work. The cavity A is in communication with the steam space E, the supply of steam being controlled by the disc valve F, which is operated by the brake handle. The space E is directly open to the boiler from which it receives its steam.

The brake is wholly manipulated by the brake handle (not shown), which forms part of the casting G, in which there are a multitude of small holes for the inlet of air when the brake is applied. There are three principal positions for the handle, namely, running position, brake on, quick release. In the second of these the handle is turned into such a position as to admit air to the main train pipe, and at the same time to shut off the supply of boiler steam to the cavity A. The third position necessitates a reversed movement of the handle, while in the first position, named above, provision has to be made for the extraction of any leakage which may take place in the apparatus. This is done by always keeping the small cavity Q in communication with the boiler, so that a jet of steam will continually flow through the annular orifice leading out of Q, and so induce the leakage to flow into the region of low-pressure at K. The valve L prevents the return of any of the exhausted air. The chamber P is in communication with the main train pipe by means of the passage R and th handle disc G.

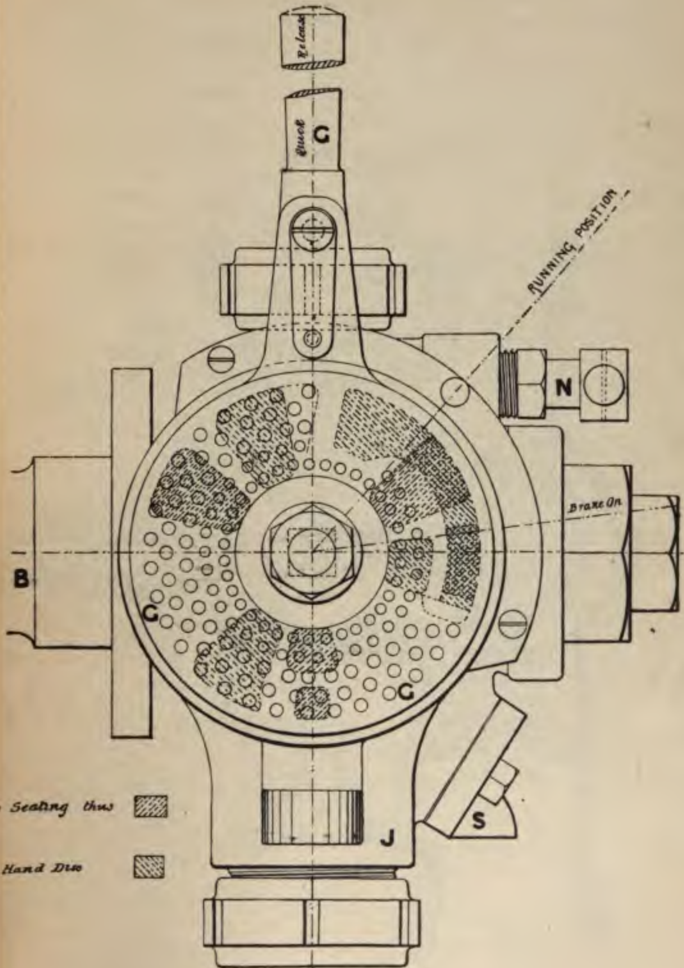

Release
Quick
G
RUNNING POSITION
N
Brake On
B
C
C
G
Seating thus
Hand Dis
J
S

FIG. 98.

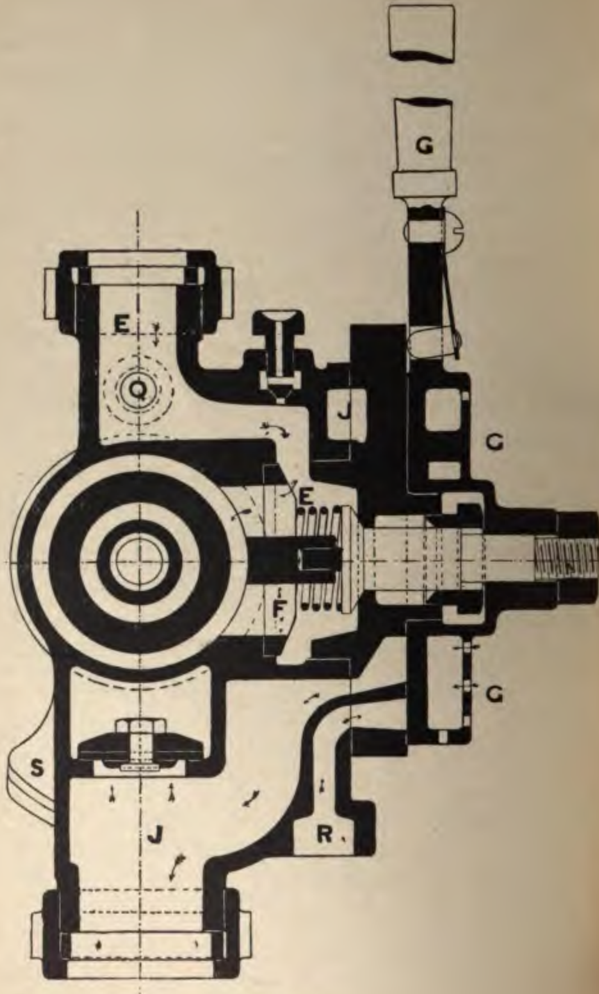


FIG. 99.

A side elevation of this ejector is given (fig. 98), with the three positions of the handle G shown. The patches, which

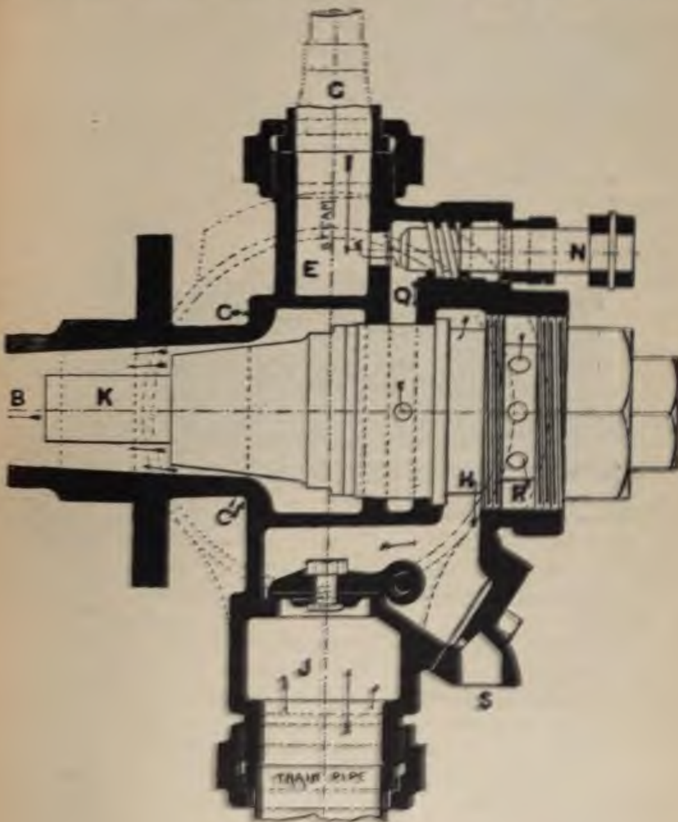


FIG. 100.

are cross-hatched from left to right *upwards*, show the dimensions of the ports in the seating underneath the handle disc; while the cross-hatching from left to right

downwards denotes the ports in the disc itself. To the flange S is attached a small drip pipe.

Fig. 99 shows a transverse vertical section. The steam passage to the large ejector is clearly shown at E, and the channel R leads from the vacuum reservoir on the engine and tender only to the small ejector. The air passage from the disc G to the air pipe is also shown by the arrows.

FIG. 101.—The Air Ejector as used by Messrs. Korting for Assisting Filtration.

A vertical longitudinal section is given in fig. 100. The valve spindle N regulates the supply of steam to the small ejector.

It is possible with this ejector to produce a vacuum of nearly 28 in. of mercury with a steam pressure of 140 lb. per square inch, but it rarely exceeds 25 in. under ordinary working conditions. On more than one railway a valve regulated by a spring is attached to the ejector to prevent the vacuum exceeding 20 in., this being the working pressure in general practice. The reason for this is that should the ejector be not able to provide so high a vacuum after a particular application that it maintained before the application, the brake blocks would be constantly pressed upon the wheels with a pressure of so many inch of mercury, and the engine would be doing useless worl besides, the longer the train, generally speaking, the mo

difficult it is to produce a high vacuum. With an engine and eighty-six carriages in a single train, a vacuum of 21 in. has been maintained, the boiler pressure being 140 lb., which seems to indicate that the length of train is practically unlimited as regards the capability of maintaining a working vacuum. A vacuum of 17 in. has also been maintained on a train of fourteen carriages, when the steam pressure was 70 lb., and 18 in. can be obtained on the engine alone with steam at 60 lb. All grease and deposit on the cones have to be removed periodically, and this can be done with an ordinary penknife.

CHAPTER XIII.

Historical Summary.

The preceding chapters have been more especially devoted to the study of the boiler-feeding injector, and it is principally the devolopment of this apparatus which we now propose to trace out in these few concluding paragraphs.

First made for commercial purposes in 1859 by the inventor, M. Jacques Giffard, in France, it soon found its way into this country, under the championship of Mr. John Robinson, managing director of the firm of Sharp, Stewart, and Co. This firm secured the patent rights for Great Britain, and began to manufacture the injector to patterns obtained from M. Giffard, while at the same time they set about developing new ones from experiments carried out by themselves, the results of which were placed before the Institution of Mechanical Engineers by Mr. Robinson, in January, 1860.

The novelty of the apparatus, and the then apparent mystery which seemed to surround its mode of action, account for the temerity exhibited in the discussion of the paper, in which only such veterans as Sir Frederick Bramwell and the late Sir William Siemens took part. This may not have been altogether unexpected, if we judge from the concluding paragraph of the paper in question, which runs thus: "It has not been attempted to give any calculations of the power obtained by the injector, and,

indeed, the writer has been discouraged from attempting this by the opinion expressed to him by an eminent hydraulic engineer, that the injector is a valuable application of a force which very few persons understand, and which has never been explained in books; and when it was found possible with steam of 24 lb. pressure to inject water into a boiler at 48 lb. pressure, it was felt that it would be premature to bring forward calculations based upon the result of experiments so hastily made, which require much consideration and discussion before any safe conclusions can be arrived at."

From a perusal of the above paper, and of a supplementary paper of a few months' later date, we gather that the author had not then become acquainted with the laws governing the evaporation of water or condensation of steam; for we find on page 75* that "great surprise was evinced at the remarkable uniformity in the increase of temperature of the feed water in passing through the injector; the average rise being 75 deg., and the extent of the variation being only between 71 deg. and 86 deg., with a range of pressure from 15 lb. to 51 lb. per square inch, and a change of initial temperature of feed from 74 deg. to 110 deg."†

It was left for Sir William Siemens to point out in the discussion for the first time that the theory of the mode of action could be easily investigated by ascertaining whether the quantity of steam condensed in the jet was sufficient to impart to the jet of water the velocity required for enabling it to overcome the resistance opposed to its entrance into the boiler; for he says: "The velocity imparted would be inversely proportionate to the weights in motion, if there were no loss through friction or eddies—*i.e.*, if 1 lb. of steam, moving at the rate of 1,200 ft. per second, impinged on 11 lb. of water at rest, the result would be $(11 + 1)$, or 12 lb. of mixed water and steam, moving at the rate of

$$\frac{1200 + 0}{11 + 1} = 100 \text{ ft. per second.}$$

* Minutes of Proceedings, Mechanical Engineers, 1860.

† These pressures appear to have been measured by an ordinary gauge, and are therefore above the atmosphere. — W. W. F. P.

This, of course, must follow, because the total momentum before impact equals the total momentum after impact.

It is interesting to find, on reference to the above paper, that the proportions of the lifting pattern used at the

FIG. 102.

present time are similar to those used by M. Giffard when he first placed the apparatus on the market, except that we do not now require the little "peephole" directly opposite the overflow gap to satisfy ourselves that the injector is at work. This fact seems to indicate that the inventor

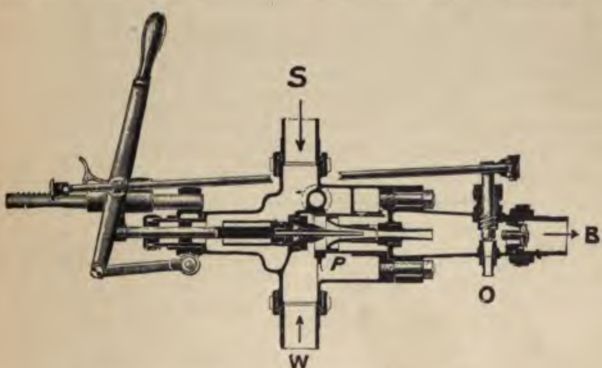


FIG. 103.—The present form of the Self-adjusting Injector.

carefully perfected his invention before offering it to the engineering public, which precaution no doubt materially promoted its adoption.

The first modification of the original "Giffard" in the way of simplification seems to have been carried out by

Messrs. Gresham and Robinson, in 1864, by making the combining and delivery cones in one casting, and operating it by a rack and pinion arrangement.

About this time Mr. Sellers, of the United States, made an alteration in the original apparatus, the details of which are shown in fig. 102. It is the first attempt that we find to supply automatic regulation. The combining and delivery cone, instead of being operated by means of a rack and pinion, have at the feed-water end a piston sliding to an air-tight fit in a metallic cylinder. The usual overflow orifice opens into a chamber, the upper end of which is covered by the above piston. If the supply of feed water is too great, some will escape by the little overflow into the chamber, causing an increase of pressure there, thereby raising the piston and reducing the water supply. Should the water supply be too little, some of the water in the chamber will be induced to flow into the delivery cone through the overflow, and thus reduce the pressure in the chamber, allowing the piston to descend and admit more feed water. The hollow steam spindle first makes its appearance in this injector for the purpose of lifting the feed water at starting; and the first relief valve in the delivery pipe is also contained in this instrument for the same purpose. This injector always delivered the maximum quantity of feed water on account of its automatic water regulation. There does not appear to have been many of these instruments made and used, though the regulation was of the most sensitive nature.

Following soon after this, in 1867, Mr. Alexander Morton, of Glasgow, now senior patrner in the firm of Morton and Thomson, obtained letters patent No. 2106, for "Improvements in the Lateral Action or Induction of Fluids, and in the Apparatus employed therefor," which patent is of historical interest, as it includes a description of the exhaust injector and the ejector condenser for the first time.

Mr. Morton begins his specification by claiming that his invention consists mainly in the construction and arrangement of apparatus by which certain currents of either elastic or liquid fluids or of gases may communicate a

simultaneous motion to other such fluids or gases by their lateral action or induction through a tube or tubes, increasing in area in a certain ratio to the diminishing

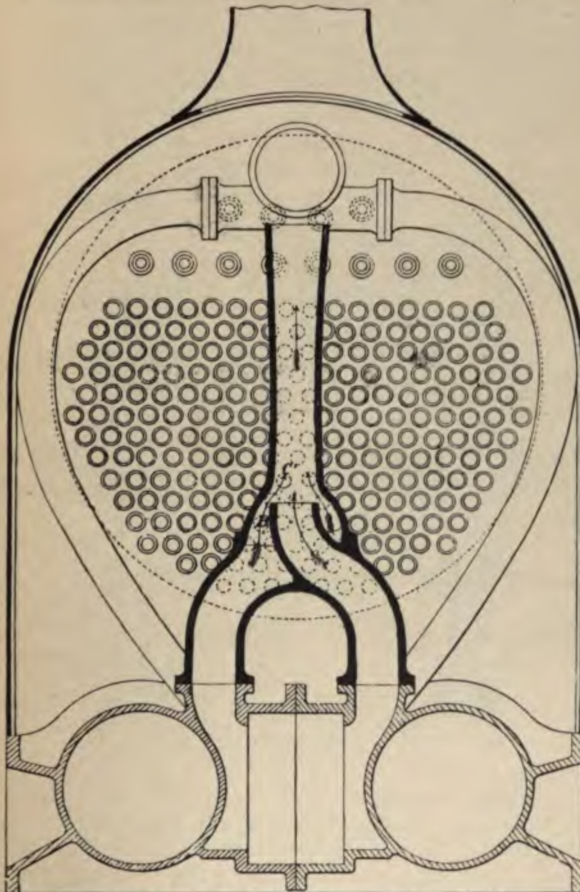


Fig. 104.

pressure of the escaping or acting fluid; and the objects and advantages to be gained by the use of the said invention are its application to steam engines and injectors, whereby such steam engines and injectors are made more perfect and economical in their working than heretofore.

The first application which seems to have come to his mind was that shown in fig. 104, where the exhaust steam

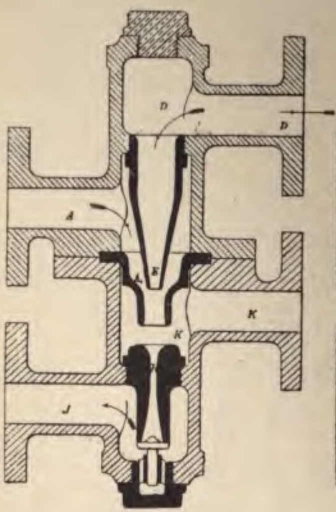


Fig. 105.

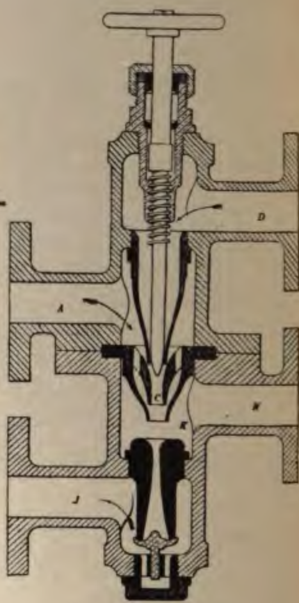


Fig. 106.

coming from each cylinder of a locomotive was so constrained as to induce some of the steam on the back stroke to flow out of the cylinder, on a principle similar to the modern method of exhausting the train pipes of the vacuum brake, and thus reduce the back pressure on the piston.

We next gather from the specification that in the place of : inducing current of exhaust steam we may, if we choose, ostitute one of water at ordinary temperatures, flowing o the apparatus under a small head. In this arrange-nt, says the patentee, "a full vacuum may be obtained h a very low head of water." In the accompanying sheet drawings there are two figures (sections) of the above paratus which are almost identical with the ejector

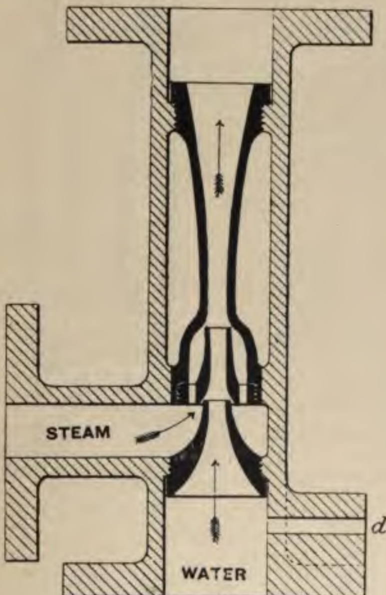


FIG. 107.

idenser now designed by Messrs. Morton and Thomson, own in figs. 87 and 88. See pages 141 and 142. Also mediately following the above we find it stated "that flow of exhaust steam into the ejector maintains and ists the jet of condensing water as it passes through the paratus, and that the water jet may be at the first

produced by a jet of live steam from the boiler, until the action is fairly started, when the exhaust steam continues the action alone."

We next find a description of an injector by which steam at a very low pressure may be made to force water into a boiler having a very high pressure of steam, in which the

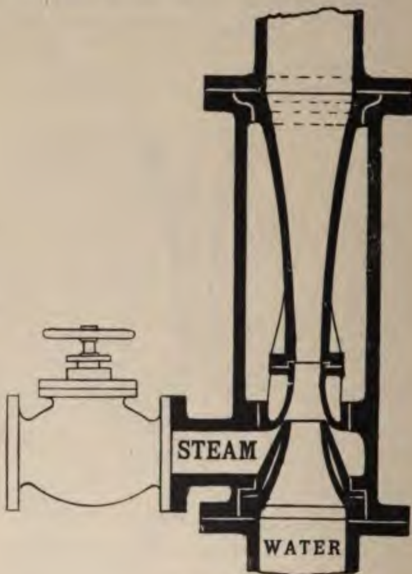


FIG. 108.

actuating steam may have performed a previous purpose before entering the apparatus. Two views of this low-pressure or exhaust injector are shown in figs. 105 and 106, the steam entering by the channel A and the water through the channel D and the central nozzle E. The inventor then goes on to say that "it will be understood, therefore, that in this improved construction of injector the water usually lost at the overflow K is saved, and that by enlarging the area of the annular jet at A, and at the same time

diminishing that of the throat H, an apparatus could be made to produce an enormous pressure and velocity of the liquid with a very low pressure of steam."

The same specification also includes the steam jet pump (fig. 107) commonly called a water lifter, whose principle is identical with that of the injector. For comparison, we may also give (fig. 108) a section of a modern water lifter made by Mr. Morton, and it will be seen that the difference between them is almost inappreciable.

In 1875, a patent was obtained by Mr. Hamer and the brothers Metcalfe for improvements in injectors actuated by exhaust steam, in which both live and exhaust steam were used in an instrument very similar to the original Giffard pattern, a lip being formed in the blast pipe, near the pipe leading to the injector, to catch the steam as it came from the cylinders. The exhaust steam was used after the live steam.

In the following year the same patentees, in conjunction with Mr. Davies, obtained another patent for the use of exhaust steam only, the apparatus being a very close copy of the original Giffard, with an enlarged steam cone.

Mr. Friedmann, of Vienna, whose name has been associated with injectors for many years, was the next to bring out an improvement, which consisted of a double water cone, which separated the water jet into two distinct annular streams, for the purpose, it is said, of allowing of the use of warmer feed water than the then injectors were able to take. One of these injectors is shown in fig. 109.

In 1876 the inventor blocked up the water to the second water cone, forming a closed chamber *r*, fig. 110, surrounding the first water cone, and drilled a hole O in the combining nozzle, which hole was put into communication with the overflow pipe *v*. By this means, he states that the apparatus is able to take hotter feed water even than before, which circumstance is due to the hole O more than to the air chamber *r*.

In the same specification, No. 4887 of 1877, we find a further and more important improvement, shown in fig. 111, though there are no reasons given for the better result obtained. It consists in putting the chamber *r* in commu-

nication with the overflow chamber *i*, and it is obvious that it is virtually the same apparatus as that previously

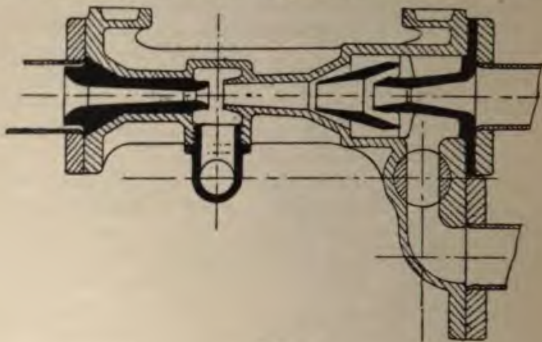


FIG. 109.

described and shown in fig. 19 *ante*, and is the precursor of all those injectors whose automaticity and re-starting capabilities depend upon a slice being removed from the

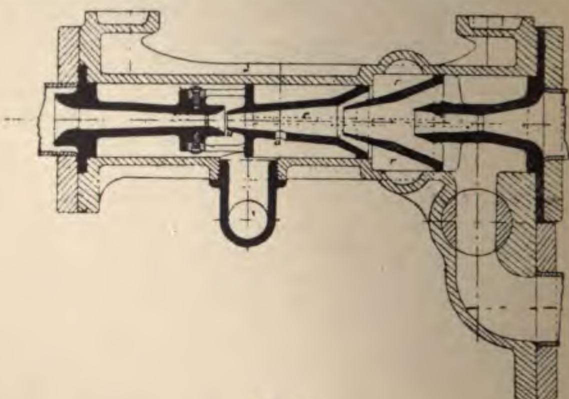


FIG. 110.

middle of the combining cone, cut by planes perpendicular to the axis, which gap provides more freedom for the exit

of the steam in starting. Of course, these injectors have an overflow valve, otherwise air would enter the combining cone and throw the injector off.

In the same year Messrs. Hamer, Metcalfe, and Davies invented the split nozzle exhaust injector, the movable portion of the combining cone being worked by hand, and altogether detached from the main body of the cone, being held tight in its place during work by a series of rods and levers, and was only used at starting. That specification

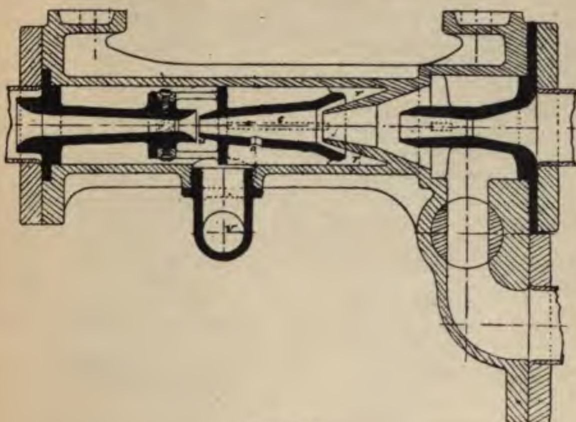


FIG. 111.

is now out of print, and hence we are not able to give an illustration of the rudimentary form of this injector, which has been further developed to so much advantage. There was no outside casing to this injector.

It was not until three years later that Mr. Davies added the hinge to the movable portion of the flap nozzle which is used at the present time. In the same patent we also find the first compound injector recorded by the Patent Office. This is given in fig. 112. Here the exhaust and live steam injectors were placed end to end, the former delivering into the latter, while the live steam for the latter was supplied through the three-way valve K and the channel

g, entering the injector in an annular jet. The hand valve K, fig. 112, also served as a relief valve in starting. A detail of the exhaust overflow is given in fig. 114. The supply of water is regulated by the hand wheel *d*, which moves the combining cone axially by means of an eccentric pin. The shape of the cones used in this injector has been maintained with hardly any modification up to the present time.

Messrs. Schäffer and Budenberg brought out a modification of the lifting injector very soon after Mr. Davies'

FIG. 112.

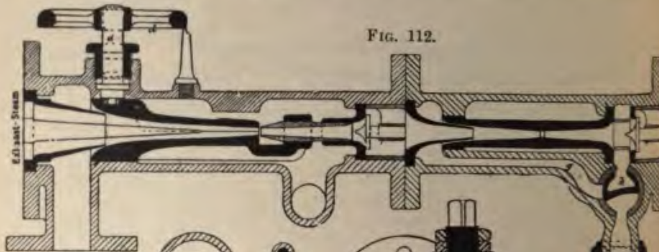


FIG. 113. FIG. 114.

compound exhaust apparatus, which had a double water channel and cone very similar in principle to the Friedmann pattern, and for the same purpose. This specification, No. 1593 of 1881, contrary to the general rule, is the essence of brevity, and merely describes the improved portion.

Almost contemporaneously Messrs. Hallum and Smith obtained a patent for the improved compound exhaust injector, shown in fig. 115, and the compound live-steam apparatus, fig. 116. The former of these, the inventors claim, can be used as a simple exhaust, a simple live-steam, or a compound exhaust apparatus. There is also some mention of its simultaneous application as an injector to aid

in producing a better vacuum in the cylinder of an engine. This last statement must be wide of the mark, as the vacuum can only be produced to an appreciable amount by passing the whole of the exhaust steam through the cones; whereas, when water has to be forced into a boiler, not

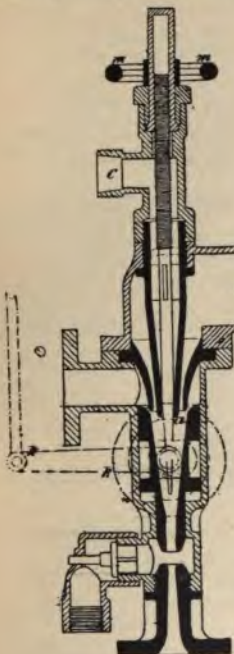


FIG. 115.

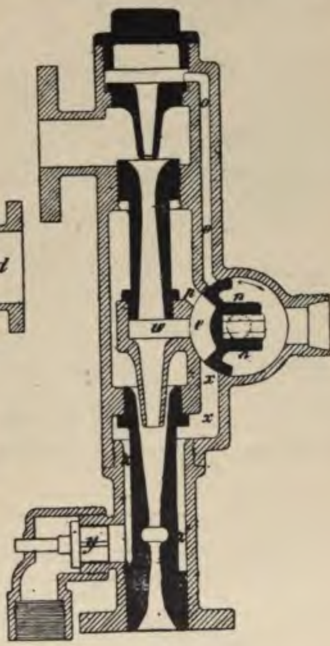


FIG. 116.

more than about one-seventh of the exhaust steam passes into the injector, and the relative dimensions of the cones for the two operations must be very different. In fig. 115 the combining cone *a* is made to slide axially by means of an eccentric and lever K, which, the inventors say, may be actuated by hand or attached to the governor. This instrument must be very inefficient as a compound exhaust

tube by the hand-wheel in fig. 117, and by the pressure in the delivery pipe in fig. 118. No overflow valve is necessary, as by the closing of the aperture no air can get into the combining cone. In fig. 119 all the cones are fixed, and the gap between the lifting tubes and combining cone closed by a couple of spherical valves. It is unnecessary to say that the form of apparatus, of which the last figure is a section, is substantially the same as that manufactured by

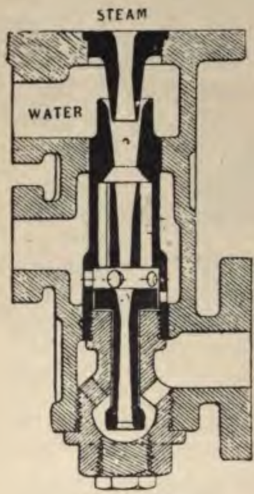


Fig. 120.

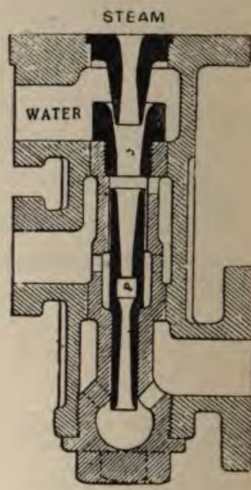


Fig. 121.

the firm to-day, for both in the early and modern apparatus there is a gap between the lifting and combining tube which communicates with a small separate chamber, the outlet to which chamber is closed by a valve or valves.

Mr. Gresham, in 1884, introduced his sliding nozzle automatic injector. Two of the original forms are shown in figs. 120 and 121, while the fourth is almost identical with that manufactured by him now, and shown in fig. 31 (see page 34). The first of these is very similar to that just *described* (*fig.* 118), the spiral spring not being used. In

the second form (fig. 121) the delivery cone only moves. On opening the steam valve it forces down the piston head and flows freely from the overflow, but on the arrival of the water the jet produces a great pressure in the delivery chamber, which pressure forces the delivery cone into its former position, closing the overflow, while the end of the delivery cone comes against the body of the injector and prevents any of the delivered water from returning to the overflow. In this apparatus the cones can be removed without breaking any joints. This system was first introduced by Mr. Gresham as far back as 1877, and is now a special feature in injectors generally. The same gentleman was the first to combine the injector with its accompanying boiler fittings, and at the same time took the steam for the vacuum brake ejector from the same casting (1885). These are what are now termed combination injectors.

Early in 1887 Messrs. Holden and Brooke adapted their "rigid nozzle" exhaust injector (which they brought out in 1884) to feed against high pressures, such as are found in locomotive practice; and in doing so their chief aim was *to render the live steam supply entirely automatic and independent of the driver or fireman*, so that the change from the previous live steam to the exhaust injector would not entail an increase of movements at starting, or an increase of attention. They have shown a number of different ways in which this can be done, but only a few can be described here. Fig. 122 indicates one method, by which the exhaust combining cone delivers into the supplementary combining cone direct. The supplementary live steam supply, coming from the channel 14, enters in an annular jet, the final delivery taking place through the pipe 5. The overflow to the supplementary portion is closed by a valve maintained against its seating by a spring, on account of the extra pressure and temperature of the jet, which would otherwise cause it to burst. The live steam channel 14 is closed by the valve 12, to the spindle of which is fixed the piston 10. At starting, the exhaust steam and water are turned on, and when the jet has been established in the exhaust portion, and the pipe 5 and its

connections become filled, the pressure it produces on the under side of the piston 10 is sufficient to lift the valve 12 against the pressure of boiler steam, and the spring thus admitting live steam to supply the additional impulse to drive the water into the boiler.

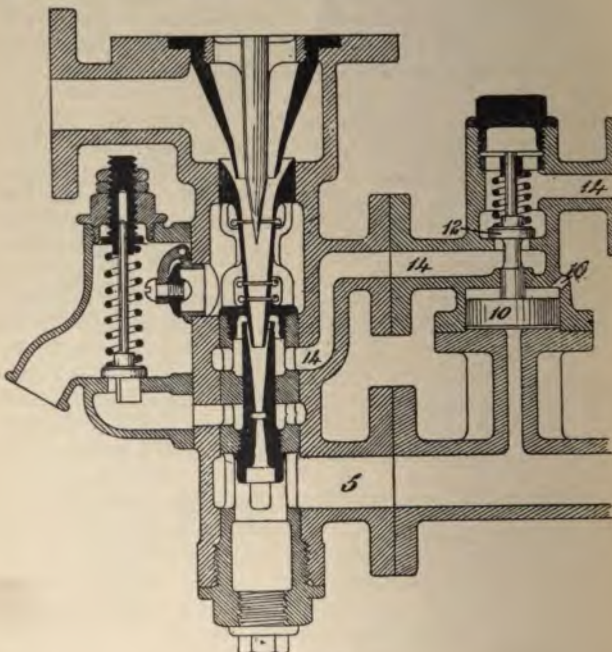


Fig. 122.

Another method was to employ a separate injector altogether for the supplementary portion, two of which are shown in figs. 123 and 124. In the former of these the delivery jet from the exhaust portion passed into the supplementary part by the central nozzle, some of which would enter the small port 13, and press upwards upon the flexible diaphragm, thus lifting the live steam valve 14,

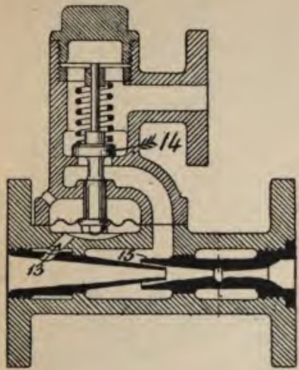


FIG. 123.

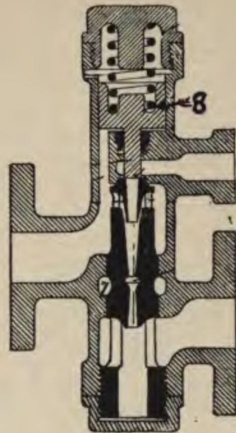


FIG. 124.

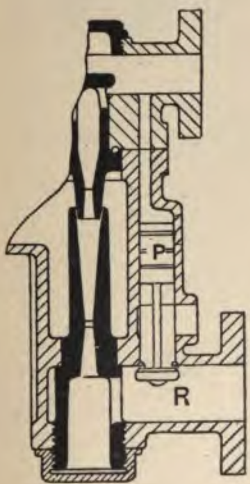


FIG. 125.

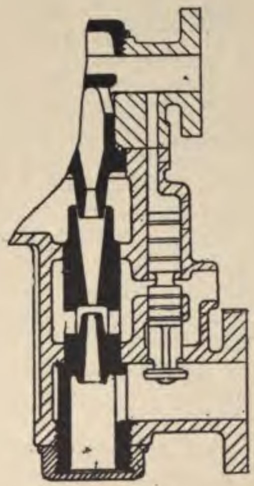


FIG. 126.

and allowing the boiler steam to enter the injector by the annular opening 15.

In the other case the exhaust delivery jet lifts the piston 8 with its valve stem, and admits live steam through the first cone; the water entering, as usual, by an annular opening between two cones.

It is in the same specification that we first find the present compact form of exhaust and supplementary portion in one casting, with their axes parallel. There are also

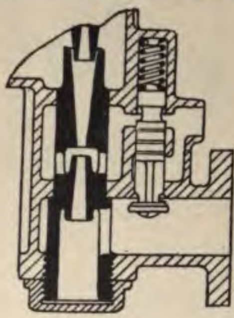


FIG. 127.

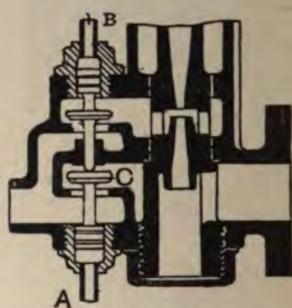


FIG. 128.

shown other arrangements, something similar to that in fig. 74, page 111, with the addition of the automatic steam valve.

The present mechanism for regulating the water and steam by the same handle at one moment is the outcome of Mr. Brooke's work in 1888, and is now used on all their live-steam injectors.

A series of different relief valves, working in connection with overflow valves, are shown in figs. 125, 126, 127, and 128, all of which are situated in the supplementary injector casting. The object of the relief valve is to assist in establishing the jet at starting or whenever the injector is thrown off, and consists of an opening in the delivery pipe, which is covered by a valve, but the valve is only closed by the application of pressure in the delivery pipe after the establishment of the jet. In fig. 125 the relief

valve R is upon the same stem as the piston P, the upper surface of which is in connection with the boiler. The boiler steam will always maintain it open until the pressure in the delivery pipe is sufficient to overcome it and close .the relief valve. Fig. 126 is a similar arrangement, but with the piston P cut in two, forming a neck which allows of the overflow being opened to the atmosphere at the same time that the relief valve opens. A spring replaces the steam pressure in fig. 127. A much more complicated arrangement is shown (fig. 128). The pipes A and B are both open to the boiler, but the former piston is larger than the latter. The two valves are of larger diameter than the pistons, so that when the injector is at work the excess of pressure upon the relief valve C will overcome that in the piston A, and the valve is closed. At the same time the higher pressure on the piston B closes the overflow valve. Should the injector be thrown off, the excess of pressure on the piston A over that on B will lift both valves, and provide a free passage for the flow of steam.

The ejector condenser seems to have thrived only in the hands of a few makers, notably Messrs. Morton and Thomson, the original patentees, and Messrs. Körting Brothers, until the energetic firm of which Mr. Ledward is the head took the matter up. The result was that in 1890 Mr. Thos. Ledward obtained, amongst other things, a patent for steam nozzles of special construction, and another in 1892 for a variable capacity condenser. The subjects of these two patents are shown in detail in figs. 129 and 130 respectively. In the former of these the water enters at the topmost flange, and passes through a grating formed in the flange of the main converging water cone, and is then delivered by the water cone into a series of co-axial steam cones, which are in communication with the exhaust steam pipe, the water stream and the condensed steam it has picked up being finally discharged through a long, diverging cone. The diverging cone is fixed into the lower portion of the ejector casing, and then the perforated sleeve fitted directly over its upper extremity. The steam cones are then placed in position one by one, their vertical ribs maintaining them co-axial

with one another, and also equidistant one from another. The last steam cone being inserted, the converging water cone is then placed in position, thus firmly securing the whole. In all of Messrs. Ledward's condensers are to be found the steam nozzles here shown with vertical ribs. The exhaust steam in all such condensers must approach the stream of condensing water at a given inclination,

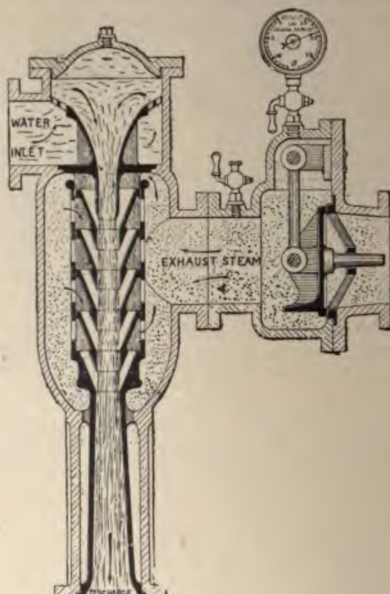


FIG. 129.

according to the shape of the cones. The stream of exhaust steam coming from the engine into the steam chamber of the condenser will necessarily contain eddies, which may distort the water jet in passing through the steam nozzles, or perhaps give the water particles a motion of rotation round the axis of the ejector, as well as a motion of translation along the axis, thus producing in the water

jet a sort of screw or tortuous motion. Now, as the total energy in the exhaust steam is only a certain limited amount, if a certain portion of it is used in producing motion of rotation, there remains less energy to produce the linear motion which is required to discharge the jet against the atmospheric pressure, and in this way, therefore, the action may be weakened to such an extent that the condenser may give out. Again, the motion of rotation of itself also tends to break up the jet and produce extra

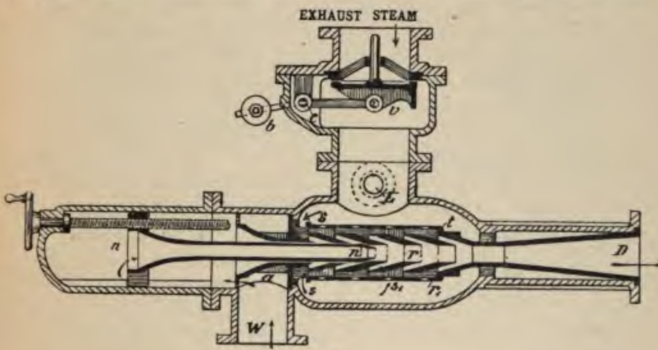


FIG. 130.

pressure where as little as possible is required. The ribs cast upon the nozzles prevent this distortion of the water jet, and therefore tend to the preservation of its action. The perforated sleeve surrounding the cones is for precisely the same object.

It will be noticed that the steam space between the cones is very large, so that any fragments of packing or sediment which may be carried along with the steam cannot choke up a portion of the space, and thus allow of the free action of the steam on only a portion of the circumference of the water jet, which would tend to distort it. This sort of thing is much more likely to occur when the steam has to pass through sets of small inclined holes, which perform the functions of the cones. This firm has paid great attention to this, and a considerable amount of experiment and

experience has strongly influenced them to adopt the particular nozzle now supplied with their condenser.

Now, as was previously mentioned, when the condenser is supplied with cold water from a source whose level or equivalent head is not less than about 15 ft., the ejector will continue to work with a very considerable range in the amount of exhaust steam supplied to the condenser, from which we gather that the velocity of the water alone, due to this head, is sufficient to produce a fairly good vacuum without deriving much aid from the exhaust steam—a circumstance which was pointed out mathematically when dealing with ejector condensers. Should the head of condensing water be much less than 15 ft. on the average (Messrs. Ledward have obtained a vacuum of $28\frac{1}{2}$ in. with a head of 12 ft.), it must either be supplied under pressure to the condenser, or the requisite additional velocity must be supplied to the water by the impulse of the exhaust steam itself. The latter method is the one generally adopted. A nearly perfect vacuum exists in the water jet itself while passing through the steam cones, and hence the supply of water to the condenser will be approximately constant. Suppose that for a time less than the normal supply of exhaust comes from the engine. The diminution in the steam with constant water supply, of course, renders the ratio of water to steam much greater, and therefore increases the vacuum. Now, Mr. Ledward has found that as the vacuum is continually increased by diminution of steam, there comes a point above which the vacuum cannot rise, while the exhaust steam maintains a continuous flow of the water jet, when the ejector is depending entirely upon the exhaust steam to keep up the velocity of the jet. This point is with pure steam at a vacuum of 27 in., or a little more with a high barometer. At this point the steam is so attenuated that its impulse is insufficient to keep up the requisite velocity of the jet, and the result is that the amospheric pressure on the discharge end prevails and forces back the water into the condenser, stopping the action altogether. Now, by reducing the surface of the water jet exposed to the steam, the condensing power of the jet is reduced, and therefore the vacuum will also be

reduced, and it is possible in this way to so accommodate the condensing surface to the amount of exhaust steam that the vacuum will be prevented from rising to such a degree as would render the apparatus inactive. If there is much air mixed with the steam, the critical point is lowered from

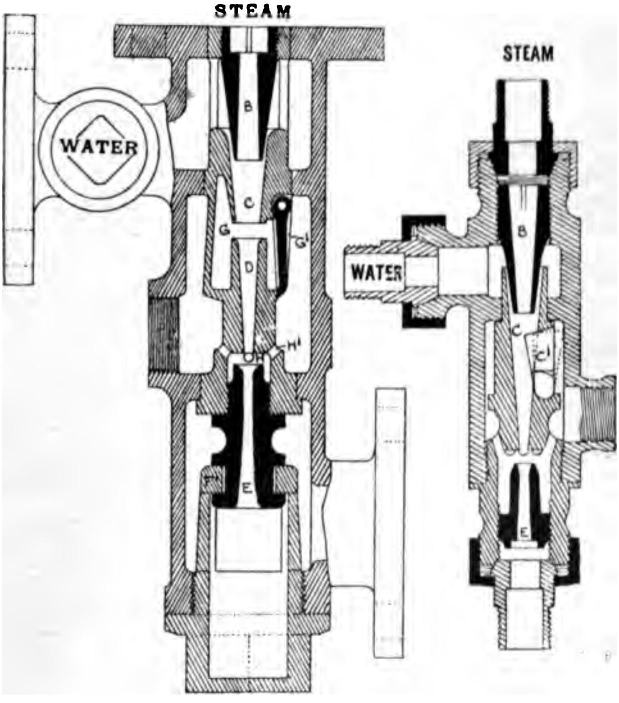


FIG. 131. FIG. 132.

27 in. to 26 in. In practice it is rarely attempted to get more than a working vacuum of 24 in., and hence a non-adjustable drowned condenser may have its supply of steam reduced about one-half without the apparatus ceasing to work.

To get over the difficulty when the head of condensing water is small, or nothing at all, the adjustable condenser is used, that one shown in fig. 130 being the invention of Mr. Ledward. The central water cone and tube *n n* is made adjustable by the hand-wheel and screw, so that any portion of the whole number of screw cones may be rendered inoperative by screwing down the tube. It is claimed for this particular apparatus that the inside of the steam orifices are nearer the jet of water than in that of any other design, so that when drawn out to its full capacity the long stream of water receives more support, and is therefore less liable to become distorted than in other systems. Another feature in the condensers, figs. 129 and 130, is the balanced non-return valve *v*, which prevents the cylinder from becoming flooded when the engine stops. The balance weight *b* just keeps the valve against its seating when not at work. Should the condenser be difficult to start, live steam may be turned into the condenser through the opening L.

Following on with his improvements in exhaust and re-starting injectors, Mr. Brooke, in 1891, adapted the latest type of his regulating "Influx" injector, to be fitted to the back firebox plate, thus forming the combination pattern.

At the same time Mr. Hopkinson, of Nottingham, patented his form of automatic injector, shown in figs. 131, 132, and 133, which in principle is identical with that of Holden and Brooke, and first patented by Mr. Brooke in 1883; but the present improvement consists in casting the walls of the chamber G together with the lifting table C and the combining tube D all in one piece, so that they may be withdrawn with the valve plate for inspection or repairs without removing the outer casing. In fig. 132 C_1 is a loose portion of the combining cone, the top end of which can fall back against the outer casing, forming a gap in the cone. The piece C_1 has its lower end partially cylindrical, and fits into a socket in the cone flange opposite the overflow pipe, which acts the same part as a hinge. This improvement appears to the author to be nearly identical with Davies and Metcalfe's flap nozzle.

Mr. Hopkinson's improvement applied to the exhaust injector is shown fig. 133. Besides the slit G we have two rings of holes *d* in the combining tube, all covered by the valve plate G_2. There are also ordinary overflow holes *h*.

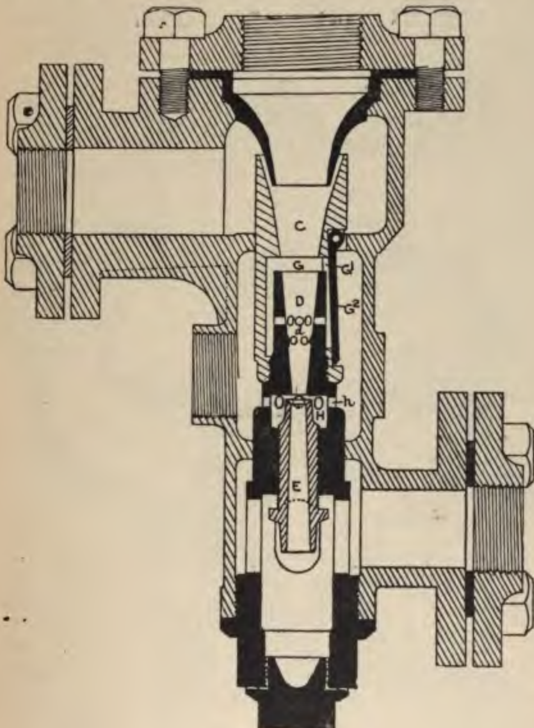


Fig. 133.

Quite recently a patent has been taken out in the United States for an almost identically similar apparatus to Holden and Brooke's "Influx" injector by Mr. W. Penberthy, a section of which is shown in fig. 134. There are two overflow valves, one covering the aperture of the supplementary

overflow, and the other the ordinary overflow. They may be both flap or spherical valves, or one of each, as shown in the figure. The number of the American patent is 484044, of April, 1891.

Another modification of the original "Influx" injector is shown in fig. 135, and has been patented so recently as

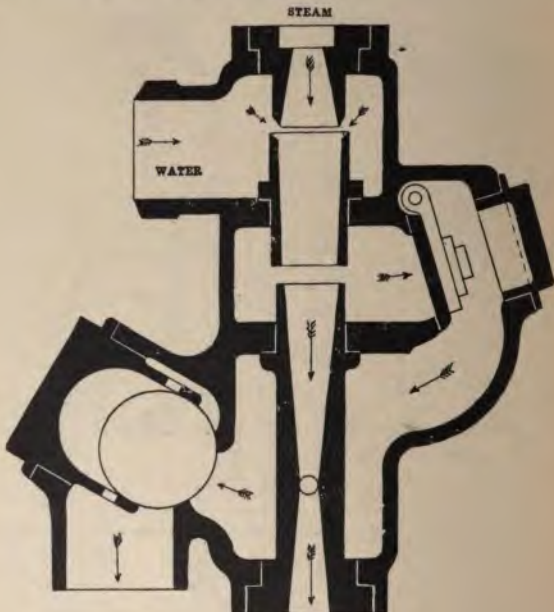


Fig. 134.

December, 1892, by Mr. O. Lindeman, of London. His modification consists of a double steam cone, the outer one receiving its steam from the inner one by means of small holes at A. The inner cone is made to extend just beyond the lower end of the lifting tube, while the outer cone is quite similar to any ordinary steam cone. The combination of parts really forms a compound injector, and if the

nary overflow pipe were closed by a loaded valve, it
ıld be capable of delivering water at a higher tempera-
than 212 deg. Fah. The annular jet of steam which
eeds from the outer cone passes through the lifting tube
escapes at the aperture D, and thence past the valve

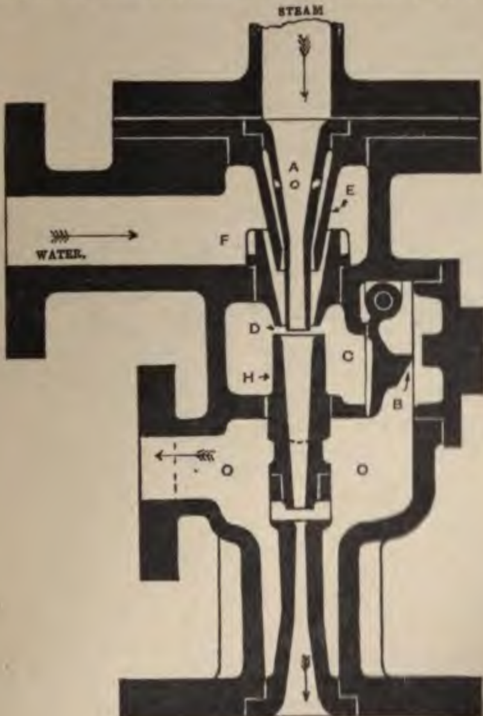


Fig. 135.

nd out at the overflow O, until the feed water arriving
condenses the steam, when the jet of water delivered
the lifting tube, and already possesses momentum by
ue of the annular steam jet, is further assisted by the
m which issues from the inner nozzle and driven into

the boiler. While the annular jet is sucking up the water the jet from the inner nozzle passes out harmlessly through the slit D and the ordinary overflow. The above injector is also, to a certain extent, a self-regulator, the amount of water passing through it depending upon how much is propelled through the lifting tube by the annular steam jet; so that if the pressure of steam falls the amount of water supplied by the annular jet also decreases.

Almost within three months of the sealing of the above patent another was granted to Mr. Friedmann for a very similar apparatus, shown in section, fig. 136. The perfora-

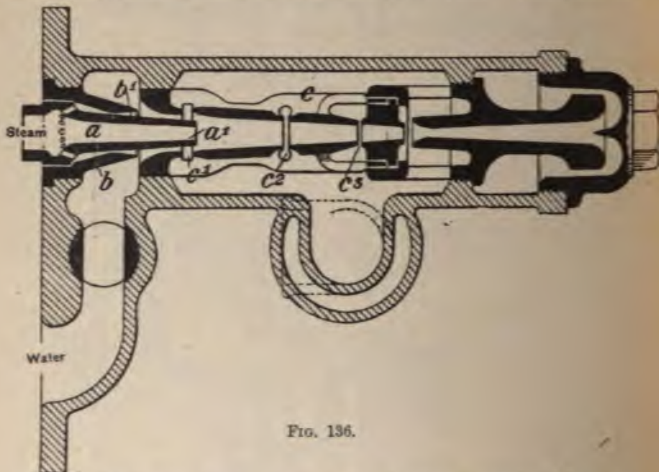


FIG. 136.

tions in the inner steam cone point in the opposite direction to those in the apparatus just described. The lifting tube is exceedingly short, and the combining cone contains a couple of transverse slits c_2 and c_3 in addition to the ordinary overflow. The patentee specifies that the sum of the areas of the openings c_1, c_2, and c_3 must be equal to the sum of the areas of the openings a_1 and b_1. The author thinks that the injector would start much more freely if the sum of the former openings were greater than the sum of the latter, the additional area being added to the

openings c_1. The several portions of the combining cone are cast together, the ribs C maintaining them quite rigid. The overflow pipe has a non-return valve in it which is not shown in the sketch; neither is the delivery flange shown. Both this specification and the last-mentioned are commendably brief.

Besides the actual feeding of boilers there are many other purposes to which the steam jet may be applied with very considerable advantage, while not only the steam jet, but the compressed air jet also, as applied to the removal of grain from one granary to another or from the hold of a ship to the granary.

The sanding of the rail has been rendered much more complete and efficient since the introduction of the steam jet to the sand pipe by Mr. Gresham, and the sand is no longer distributed over the track instead of on the rail face. A section of the apparatus is given in fig. 137, which also contains the steam valve in transverse and longitudinal section. This valve is so constructed that when not supplying steam to the sanding ejector, any moisture which may leak through the valve is led away down a drip pipe. The jet of steam passing through the ejector induces a current of air down the large pipe joining it to the sand trap, which air enters the hood over the trap entrance by the inlet A, and then follows the arrows in their curvilinear path, at the same time licking up some sand just before descending towards the ejector. When the sand stream arrives at the steam nozzle it is driven forward by the jet down the inclined pipe, which is so directed that the sand will impinge on the rail just in front of the wheel. The incline iron pipe gets worn away by the sand blast, and has to be frequently renewed.

This apparatus has materially enhanced the value of locomotives with single pairs of driving wheels, allowing of very much increased adhesion at starting and in ascending inclines. The pull on the drawbar, on the average, during a long run is small compared with what it is at starting, and hence the increase of adhesion accomplished by the sand blast makes a simple driver almost equal to a four-wheel coupled engine.

combustion of the fuel independent of that drawn through the grate and fire beneath by the action of the draught.

The appliance under notice forms an injector and ejector combined. The central inner cone is of annular formation, and acts as an exhausting device which draws air through a series of heaters placed in the smoke-box or flues, and delivers it in the middle of the oil fuel spray.

Figs. 138, 139, and 140 show the latest patterns of these burners for locomotives, and for stationary and marine use. The inlets for steam and oil fuel are marked, and the jets and spray with the induced currents of air are represented by lines and arrows.

The general arangement of the apparatus on a locomotive is shown in figs. 141, 142, and 143, to which the following reference applies:—

The oil fuel is carried in a suitable tank on the tender, and is heated as it passes to the engine by traversing a chamber kept hot by the exhaust steam from the air-brake pump. It is then conveyed to the injectors or burners through approved regulating gear, and is injected, atomised, and oxygenated by the combined jets of steam and heated air.

Figs. 144 and 145 represent the application of two of the stationary pattern "Holden" injectors to a Lancashire boiler. It will be noticed the fittings here are simpler than those applied to a locomotive, for the reason that on all stationary and marine boilers the burners, pipes, etc., are more accessible, and consequently need not be provided with the refinements necessary on locomotives, where facility of interchange, combined with perfect control of the apparatus, are essential.

Figs. 145 and 146 show the "Holden" burner applied to the furnace of a torpedo-boat boiler, and arranged to operate with forced draught on a system devised by Mr. B. H. Thwaite, as illustrated in fig. 148. He sometimes uses a rifled injector for the purpose of giving the liquid a spiral course, which aids in thoroughly mixing the fuel and air as it is drawn into the furnace. In Mr. Thwaite's arrangement, which is fitted to torpedo boats, and used more especially when full power is demanded, the oil is first sprayed by steam into the hot-air trunk of the

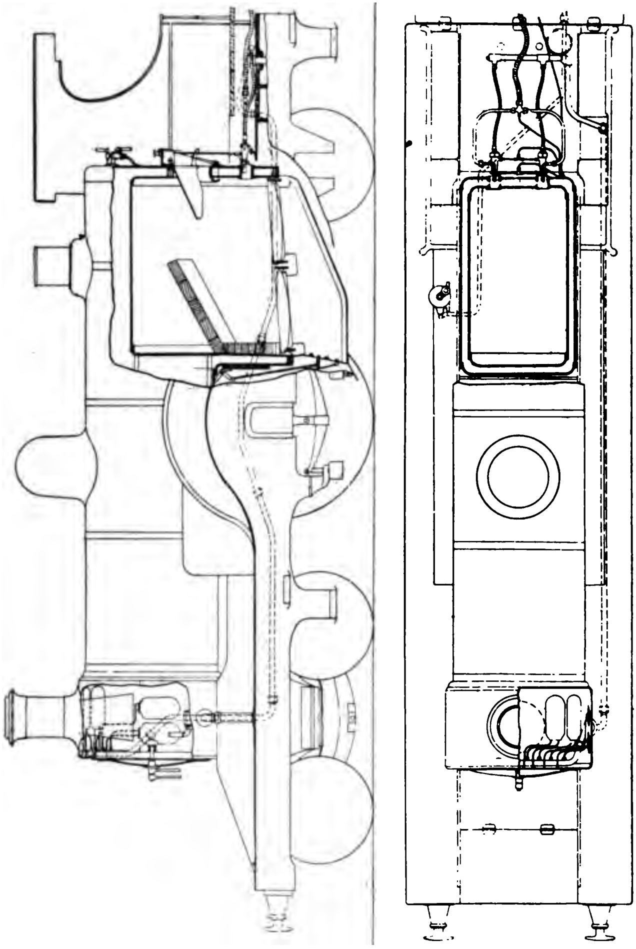


FIG. 142.—Elevation and plan of engine.

forced-draught appliance, and thus it becomes thoroughly intermixed and vaporised before it approaches the fire. Mr. Urquhart has introduced a system of oil-firing for locomotives something similar to Mr. Holden's, and used it with great success on some of the Russian railways.

Ejectors are now used for raising ashes from the stoke-holds and throwing them overboard. The apparatus is very similar to the ordinary air ejector.

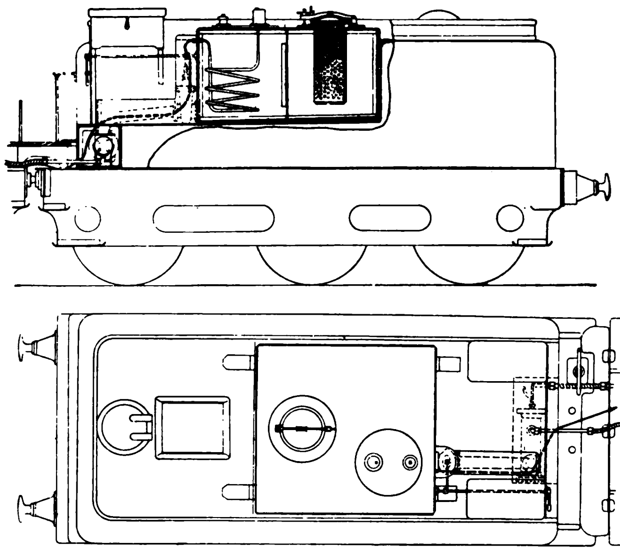


FIG. 143.—Elevation and plan of tender.

Professor Carpenter, of Sibley College, Cornell University, U.S.A., has used the injector of small dimensions to determine the amount of moisture held in suspension by steam as it passes from the boiler to the cylinder of an engine. A tank of water is weighed, and the injector, which derives its motive power steam from the main steam pipe, draws from and delivers into the same weighed tank *of water,* so that at the end of the experiment the tank

also contains a certain amount of condensed steam. By the use of accurate and delicate thermometers and pressure gauges, and the results of Regnault's classic experiments on the evaporation of water, a determination of the amount of moisture may be obtained.

Other applications of the injector principle may be cited in minute detail, but probably enough has already been given to show its vast utility.

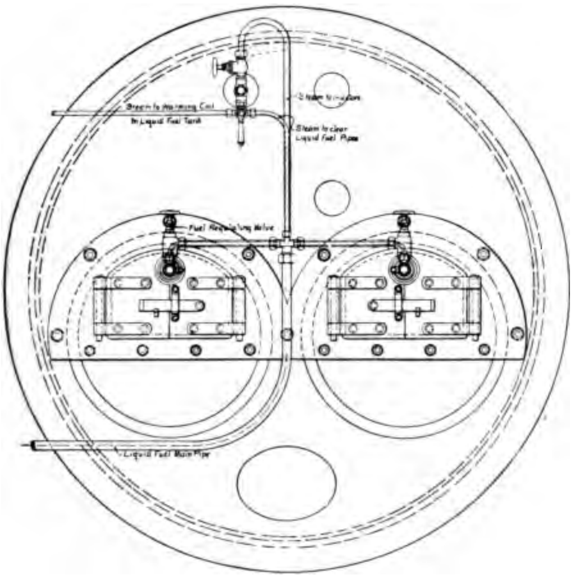


FIG. 144.

Contrary to general usage, in the previous equations dealing with the live-steam injector the author has adopted on all occasions an expression for the velocity of efflux of the steam jet, which is that which mathematical reasoning shows should be used, when the difference of pressures causing the flow were such as to produce the maximum weight of steam to flow out per second (generally termed

maximum weight-flow), when it is assumed that we are dealing with a so-called perfect gas, the conditions of flow being adiabatic.

Some excuse for this wide departure from the course generally taken needs a little comment, and hence this digression.

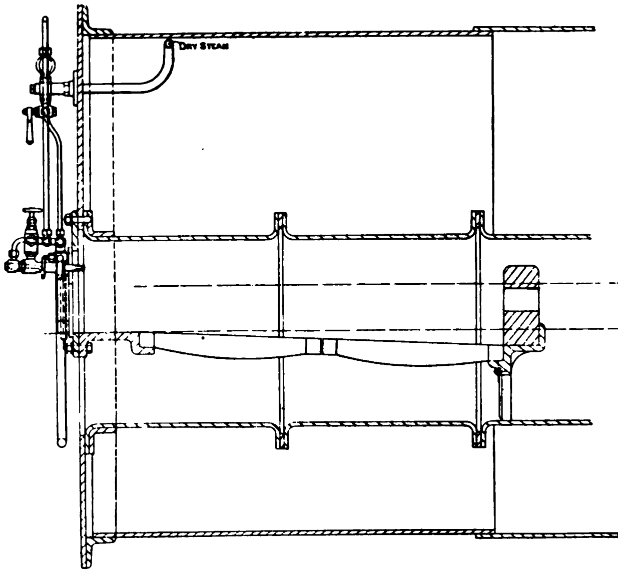


FIG. 145.

It has been previously stated that Mr. Napier found in his experiments on the efflux of steam, that with a constant initial boiler pressure the weight of steam flowing out per second increased as the final or exit pressure was decreased, until this latter was reduced to three-fifths of the boiler pressure, a result predicted by calculation. On a further decrease of the final pressure no appreciable change was found in the weight-flow of steam. Hence we may conclude that the maximum weight-flow occurs when the pressure *in the chamber* receiving the flowing steam is anything

between zero and three-fifths of the boiler pressure; or, in other words, under ordinary circumstances the weight of steam flowing out of a boiler per second to an injector will be constant. Now, the impulse of the steam jet is the change in the momentum of its material per second—*i.e.*,

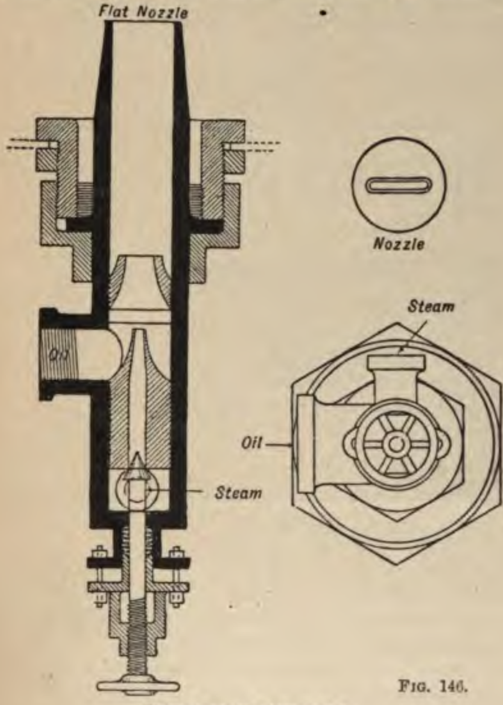
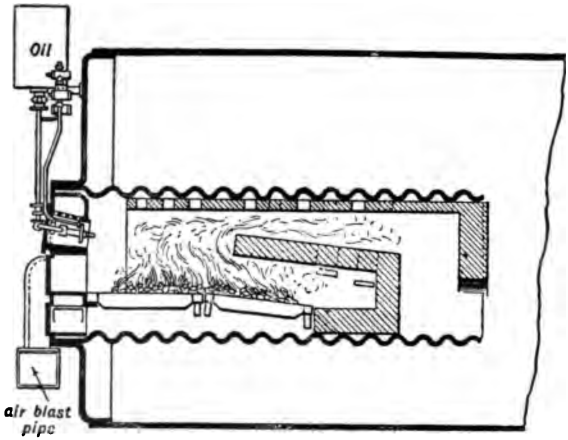


FIG. 146.

THWAITE'S INJECTOR.

the mass-flow per second by its velocity. The former factor we have shown above to be constant, and hence the impulse varies as the velocity. Again, as we have assumed for the sake of simplicity and convenience that we are dealing with a perfect gas, the velocity, according to mathematical reasoning, ought to increase as the difference of pressure

increases. But how far are we right in our assumption of the resemblance of steam to a perfect gas? The average steam to be found in a boiler is very far from being a perfect gas, or even a perfect vapour, holding, as it does, a quantity of mist or moisture in suspension. Clearly, then, we must modify the expressions for the velocity of efflux in our calculations to accord as nearly as possible with the actual results, which we can only do by a further



OIL-FIRING IN TORPEDO BOAT.

FIG. 147.

reference to experiment. These have yet to be made, but those experiments which have been already carried out upon injectors seem to indicate that the expression used by the author for the velocity of flow is not very wide of the truth, and the error is on the right side.

Some experiments on steam jets have been made by Mr. Walter Rosenhain, and an account of the results have been published in the Transactions of the Institute of Civil Engineers, vol. cxl., whilst these sheets were in the press. Mr. Rosenhain measured the momentum of a variety of steam jets by means of a special piece of apparatus, which continuously indicated the pressure of the steam in the chamber to which the jets were attached. The boiler used

was of the locomotive type, the steam pressure varied from 0 to 200 lb. per square inch, and the steam was admitted to the jet at full boiler pressure, the stop-valve being wide open. Experiments were also made using the valve partly closed, thus throttling steam of high pressure

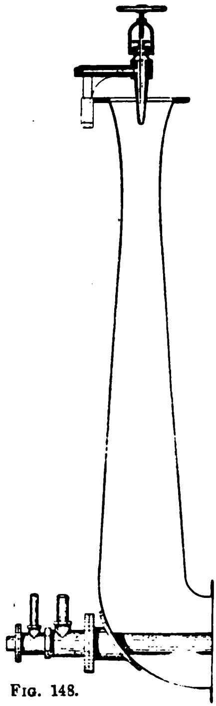


FIG. 148.

down to any lower pressure desired. So far as the reaction of the jet was concerned there was no difference between the amounts recorded by either method of obtaining the pressure in the chamber supplying the jet; but when there was much throttling, as from 200 lb. down to 20 lb., the jet was observed to be of a darker colour, much more transparent, and showed a brown colour by transmitted

light much more strongly. The weight of steam discharged per second was measured for each jet. The highest velocity attained was with steam at 200 lb. pressure, and amounted to 3,060 ft. per second. The results are plotted as curves, which are discussed, and conclusions are drawn from the experiments to guide one in the design of the most efficient nozzle—*i.e.*, a nozzle that will develop the greatest kinetic energy in the jet per pound of steam consumed.

Should the actual velocity be definitely known in any particular case, it can be substituted for that of maximum weight-flow in the proper equations.

There is another cause for the periodical inspection of steam injectors besides that of deposition of sediment, and it is "pitting." The cones, being made of brass or gun-metal, are readily attacked by water at a high temperature containing a small amount of acidity, and in time the once perfect cone becomes a mass of spongy material. Besides the destruction of the metal, we have an exceedingly rough surface, over which the feed water is continually passing at a high velocity; and hence there must be a considerable resistance offered to its motion, which tends to throw the injector off. A thin coating of sediment, of course, prevents this action of the acidulated water, though it also provides a surface which is not so smooth as the pure metal. The cones may be also cut or scored by the passage of grit, derived from impure water, such as would be found in estuaries of rivers. Another cause of stoppage is the local deposition of sediment, which produces eddies: and sometimes bad design will account for local pitting and stoppage.

In illustrating this historical summary, the author has made free use of the drawings which accompany patent specifications, and has selected those which appear to form links in the chain of development of the apparatus.

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